# 4th and 5th Axis Rotary Table

Degrees of Freedom

**Team:**
Dakota Bass
Ricky Riedl
Nicole Slagle
Irene Yee

**Sponsors:**
Martin Koch
Dr. Jose Macedo

Mechancical Engineering Department
California Polytechnic State University
San Luis Obispo
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# Executive Summary

Dr. Jose Macedo and Professor Martin Koch of the IME department saw that there was no available 5th axis rotaries compatable with smaller CNC machines, and desired one in order machine more complex wax patterns for lost wax casting. Commercial 5th axis rotaries typically cost around 30,000 USD, which is quite an investment for smaller institutions and so our sponsors Koch tasked our team with designing, building, and testing a 5th axis rotary table that will match commercial specifications at much lower, aproachable price point. The project began by researching existing rotaries and developing a list of technical specifications that the design must satisfy. Using these criteria, a preliminary design was created and initial calculations were performed to verify the design. The first prototype was manufactured and after performing a few tests, changes for improving our next generation design was determined. The design went through a complete overhaul, and a new prototype that was more robust and better designed for manufacturing was machined using CNC machines. Based of testing results, each rotary stage has a natural frequency of about 400 Hz, which should be avoided when used for machining. A 4th axis rotary table using a Haas specific Yaskawa motor was made as well as a complete 5th axis rotary table. Over the course of the last quarter, the choice of motor changed due to sponsor-related circumstances, and so the resulting 5th axis design was chosen to be driven by two stepper motor and an external controller that works in conjunction with a G-code macro.

# 1. Introduction

## 1.1 Sponsor Background

The sponsor of this project is Dr. Jose Macedo and Professor Martin Koch of the IME department. Professor Koch often machines wax for lost wax casting instead of molding the patterns, since the designs are typically unique. He and Dr. Macedo wanted to explore the possibility of designing and building a $5^{th}$ axis to machine complex patterns, but it must fit in his existing Haas Office Mill.


*Figure 1. Haas OM-2 (Haas)*

## 1.2 Formal Project Definition

The IME department would like to have five axis simultaneous machining capabilities to manufacture small wax parts. There are two Haas OM-2 vertical machining centers, which are currently available for this project. Due to their small size, there are no commercial quality 4th and 5th axis rotary tables available in the market. This team was created in order to design and build a low cost 4th and 5th axis rotary table for the OM-2.

## 1.3 Objectives

Completion of this project entails the development of a $5^{th}$ axis rotary table with compatible fixture connections, appropriate drive cards, and 5th axis CAM software and postprocessor. The table must be able to rotate in two axes, while allowing the largest work piece possible. It must mount to standard T-Slots, and have a platter with a standard bolt pattern so that commercial fixtures may be attached. Structurally, we are aiming to resist light cutting forces in wax or plastic under high-speed conditions. Our goal is to match the resolution, accuracy, and repeatability of existing large rotaries in the 25,000 USD price range. This goal is stated with the understanding that commercial systems reach their specifications by having extremely tight component tolerances, which are typically produced with dedicated machines, and may not be entirely achievable.

## 1.4 Project Management

For this project, while everyone had a part in every step of the process, individuals were assigned to manage certain aspects of the project in order to assure that the project would be completed on time. The design of the rotary was completed with every team member present. Ricky led the analysis of crucial components and Nicole was in charge of the mechatronics side of the rotary in terms of build and design. Irene managed the scheduling and the timeline to ensure that the team was on track for completion. Irene was also the main communicator with the sponsor. When time came for manufacturing, Dakota lead the manufacturing due to his background with CNC machines. The vibrations tests were conducted under Ricky and the Mechatronics tests were conducted under Nicole.

# 2. Background

## 2.1 History

Over the past two decades, the machine tool industry has been experiencing a shift toward lighter, smaller CNC machine tools (Arnold). Advancements in spindle tapers and grinding accuracy have allowed small taper machines to compete with large machines. Popular small taper standards include HSK, BT-30, and BT-30 Dual Contact spindles (All Industrial). This shift towards smaller machines has been caused by the ability of such tapers to take higher cutting forces than before. Also, high speed machining allows for the material removal rate (MRR) of these tools to match the MRR of larger machines (Albert).

The trend towards smaller CNC machines in industry has lowered the cost of small table size / small taper vertical machining centers to approximately 50,000 USD (Haas). At the same time, advances in CAM software have made programming parts simpler than ever before. More consumer groups, such as high schools, light industry, universities, and dental practices are capitalizing on the availability of these CNC machine tools.

## 2.2 Existing Products

A concurrent trend in industry has been the utilization of 5th axis rotaries. Aftermarket 5th axis rotary tables can be bought for around 30,000 USD and add two additional axes to the three existing axes of a CNC machine. This allows parts to be machined more accurately in fewer setups while enabling the manufacturing of more complex parts. In many situations, purchasing a 5th axis table can be a profitable business decision (Sherman). However, commercial quality 5th axis rotary tables are not as available for the smaller machine market. Typical manufacturers of

*Figure 2. Haas TRT 100 (Haas)*

*Figure 3. Nikken 5AX-130FA (Nikken)*

5th axis tables include Lyndex-Nikken, Haas Automation, Tsudakoma, and Matsumoto. Their 5th axis tables are not designed for machines with small table sizes, such as the Haas OM-2. Rotaries within the budget of small organizations are oversized and too heavy for smaller CNC machines. There is a desire for small 5th axis rotary tables for small table sizes and low cutting forces, which could bridge the gap between hobbyist tables and industrial tables. In the Cal Poly IME 141 lab, Haas OM-2 CNC machines are utilized to make wax shapes for sand casting. For wax and soft plastic machining, there is a niche for accurate, but small 5th axis rotaries.

## 2.3 Patents

Below are a few patents relevant to our design.

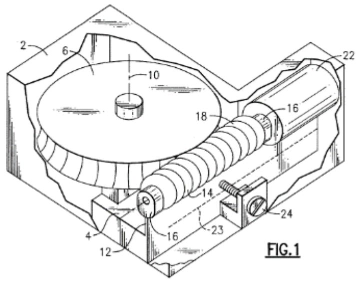



*Figure 4. Anti-Backlash Mechanism Patent Design (Mauro)*

Patent 6,016,716

This is a patent for reducing backlash in a worm gear drive. The user must manually dial in the setscrew to adjust the contact between the worm and the gear. Over time, the setscrew must be reset.

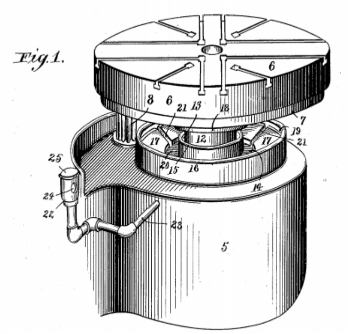



*Figure 5. Rotary Table Bearing Patent Design (Bullard)*

Patent 828,876

Bullard's rotary patent uses an external oil port to keep the rotating contact points submerged at all times. This decreases the effects of wear on the rotary over time and minimized maintenance.

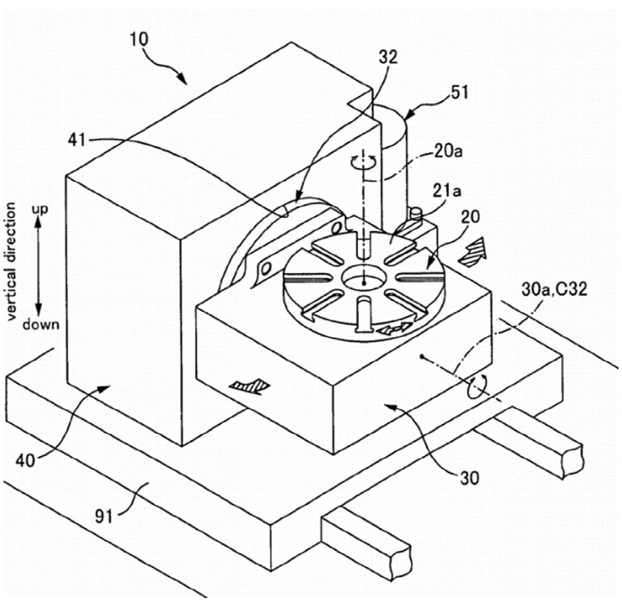



*Figure 6. Rotary Table Apparatus Patent Design (Kato)*

Patent US 7,603,930 B2

The Rotary Table Apparatus is essentially a fourth and fifth axis rotary table. This design focuses on small and light-weight properties.

## 2.4 Component Research

### 2.4.1 Backlash Reduction

In order to achieve tight tolerances while machining, it is crucial to reduce backlash within the transmission. In industry, there are several methods by which to eliminate backlash in worm drives.

**Precision Ground Gears**

Tight tolerance gears are typically used in machine tools to reduce backlash. They are finish ground using special machinery to exact tolerances. The cost of such a system is excessively high.

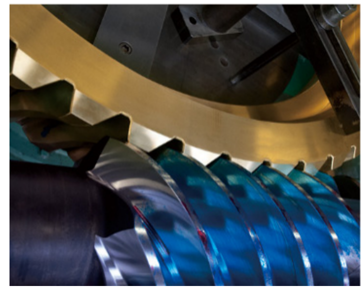


*Figure 7. Manufacturers in Japan use blue ink to machine precision ground gears (Mitzubishi)*

**Cone/Hourglass Worm**

Cone drives are custom ground gears manufactured by Cone Drive Inc. They feature a unique design that has a higher contact pattern that limits backlash. Their custom design is not COTS and therefore expensive.

*Figure 8. Cone drive manufactured by Cone Drive Solutions (Cone Drive)*

**Duplex Worm**

These worms have continuously increasing tooth thickness to insure continuing contact with adjustment in the axial direction. They require custom grinding and final manual adjustment. They also require fine tuning as the gears wear with usage.

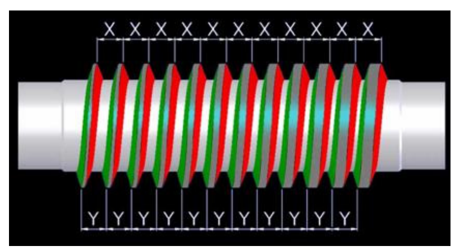


*Figure 9. Duplex worm gear drawing provided by Allytech (Allytech)*

**Anti-Backlash Split Gears**

Anti-backlash split gears are commercially available gear systems that are designed for light loads. They eliminate backlash through a spring loaded second gear that takes up the slop in the gear train. These gears are more expensive than standard gears, and are an effective solution for light loads.

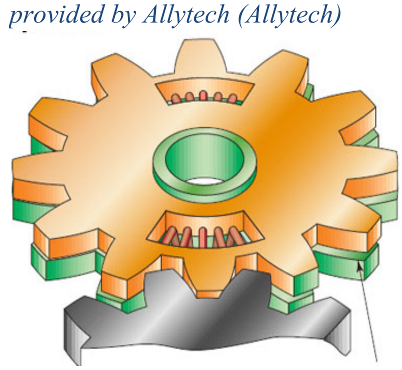


*Figure 10. Split gear (Machine Design)*

**Manually Adjusted Gear Mesh**
Manually dialing in a worm drive is common practice for manual rotary tables. The worm is adjusted via a set screw that brings the worm and gear into each other. This allows radial locational adjustment to counteract manufacturing errors. The "Anti-Backlash Mechanism for a Rotary Stage" patent by George Mauro depicts this design and is further described in the Patent section of this report.

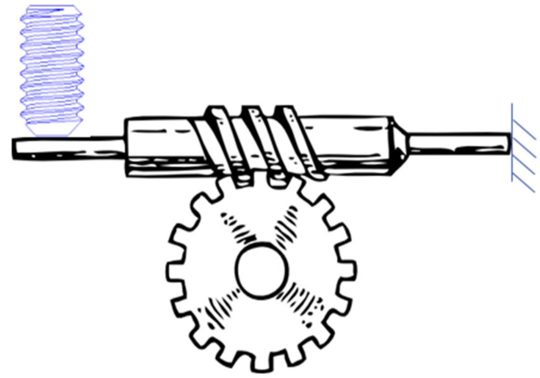


*Figure 11. Manually adjusted worm drive*

**Spring Loaded Gear Mesh**
Spring loaded gear meshes force the teeth of gears into constant contact via a spring force. This accounts for variations in diameter along the outer profile of the gears.

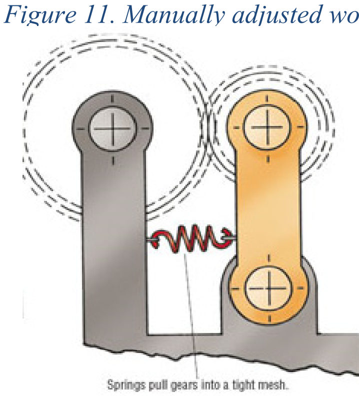



*Figure 12. Spring loaded spur gears to reduce backlash (Machine Design)*

Table 1 shows the three main criteria for backlash reduction options. As with many components, there is a balance between cost and quality that must be analyzed. Each method is suited for different types of applications and used in certain types of systems. In addition, each solution has differing amounts of backlash reduction.

*Table 1. Comparing Backlash Reduction Options (Machine Design)*

**Low-backlash gear options**

| Backlash control method | Relative cost | Most frequent use | Typical backlash |
|---|---|---|---|
| **Close mounted gears (adjustable center and spring loaded)** | Medium | Most widely used method. | About 1/4 deg. |
| **Spring loaded split gears** | Low | Light loads such as instrumentation. | Close to zero. |
| **Backloading or dual gear trains** | High | Heavy duty servo systems. Accommodates heavy loads. | Close to zero, but subject to high wear. |
| **Plastic filler** | Low | Low speeds and light loads. Light instrumentation. | No data. |
| **Tapered gears** | High | Low speeds. | No data. |
| **Preloaded gear train** | Medium | Limited rotation applications. Instruments. | Close to zero. |

### 2.4.2 Motors

Each axis must be controlled independently, which requires two motors and a controller. From collaboration with Haas Automation and Yaskawa, we were initially advised to choose among the Yaskawa SGMJV motor series to be compatible with the Haas controller. Unfortunately, months later, we found out that the SGMJV motors are not compatible with the Haas controller. Instead, we were given proprietary 200-watt motors. The rest of this section includes the design process of sizing a SGMJV motor, which were ultimately not used in this project, but useful in sizing the appropriate NEMA stepper motors and for further advancements on this project.

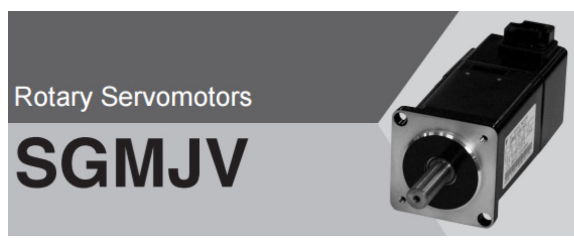



*Figure 13. SGMJV AC Servo Motor (Yaskawa)*

In order to calculate the required motor torque, the cutting force was experimentally determined. A 2 flute 3/8” endmill was used to machine wax with a .250” depth of cut in a full slotting machining operation. The machine torque was measured using a load meter and a large amount of data (32 pcs) was collected at random time intervals. The load of the spindle motor when it was not cutting was also collected to account for inertial effects.

Following this, we knew that the cutting torque was 0.25Nm as seen by the workpiece. The radius of our tool was 3.76mm, which allows us to calculate the cutting force distribution as shown below.

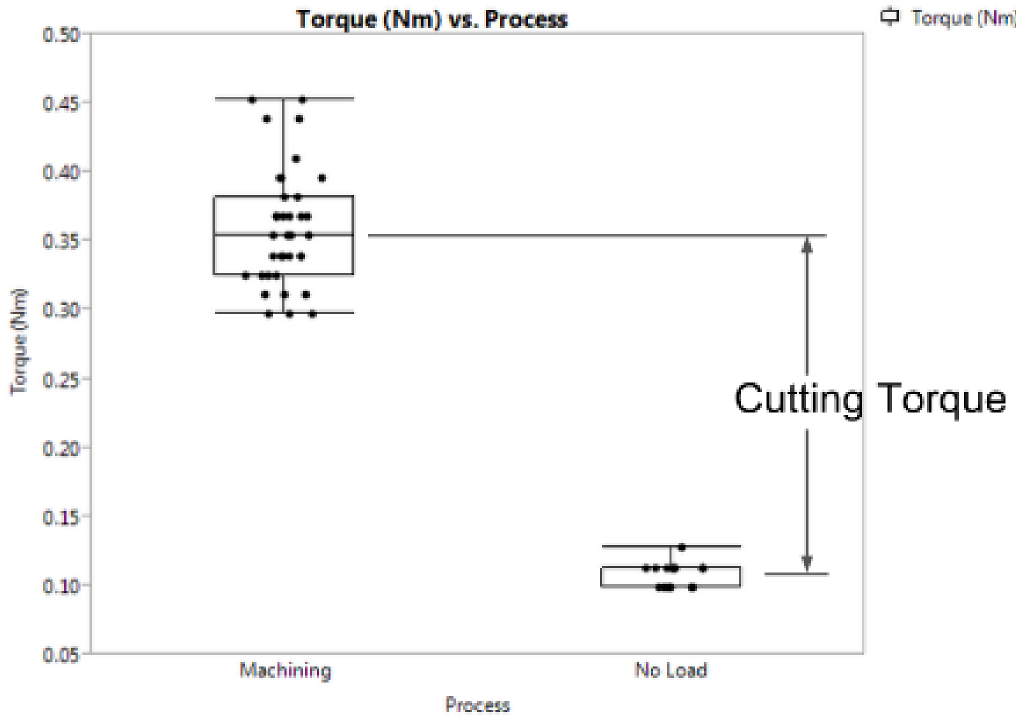
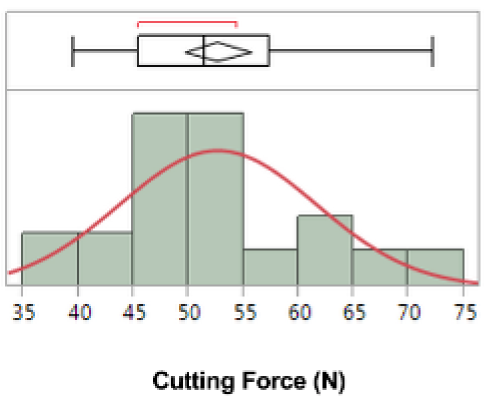



*Figure 14. Cutting Torque Data*




**Summary Statistics**

| | |
|---|---|
| Mean | 52.613101 |
| Std Dev | 8.9564167 |
| Std Err Mean | 1.4927361 |
| Upper 95% Mean | 55.643516 |
| Lower 95% Mean | 49.582685 |
| N | 36 |

*Figure 15. Cutting Force Distribution*

Assuming six standard deviations, the max cutting force is 80 Newtons. With a 100mm platter diameter, the torque seen by the gear train is 4Nm for the B axis, and 8Nm for the A axis.

In addition to cutting forces, the table velocity profile must be specified in order to size a motor. The velocity profile below shows the table accelerating to 100 rpm in 0.5 seconds and switching directions after a one-hour interval. This is considered our worst-case scenario motor operation.

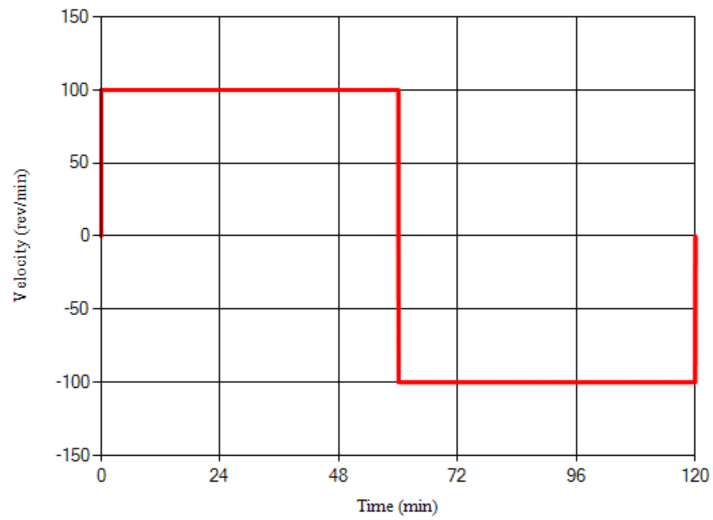



*Figure 16. Velocity Profile for Both Axes (Yaskawa)*

After determining the operating point, the SigmaSelect program offered by Yaskawa was used to size the motors. SigmaSelect inputs are applied torque, inertia, speed, efficiency, gear ratio, and velocity profile. Outputs include motor recommendations and sizing options. The following tables are inputs and outputs for the SigmaSelect program.

Furthermore, Yaskawa engineers recommended specifying an oil seal to prevent small contaminants from entering the motor. In the motor performance curves below, the dotted line refers to the 20% due to the oil seal.

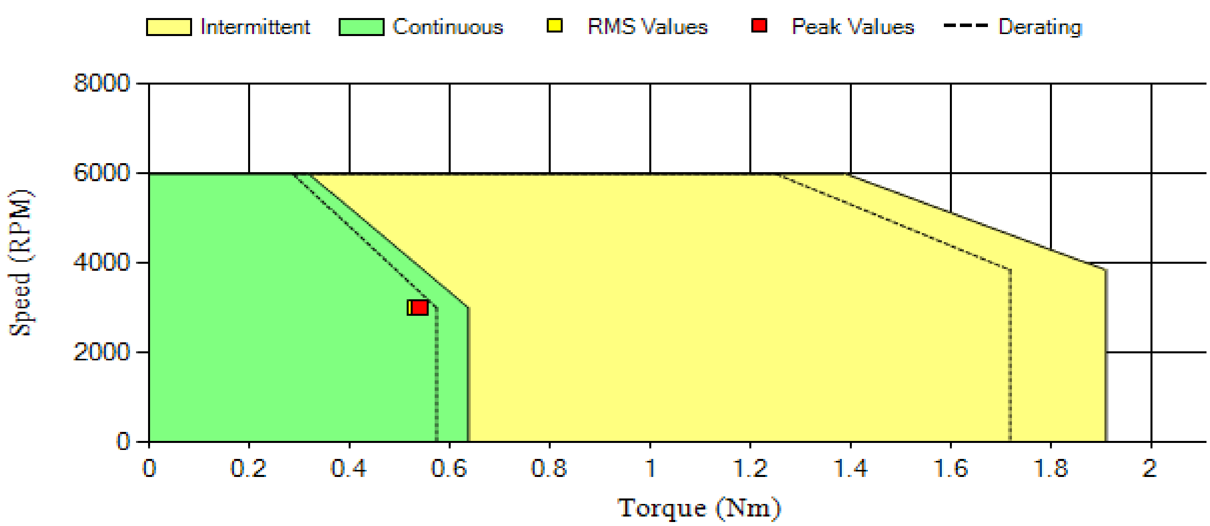



*Figure 17. Axis A Motor Performance and Operating Point*

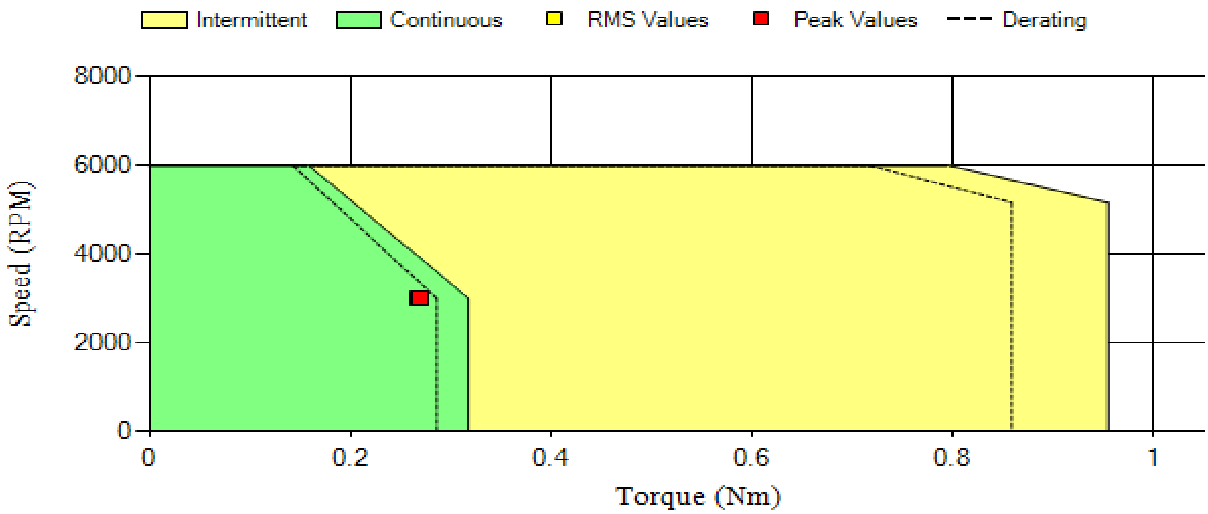



*Figure 18. Axis B Motor Performance and Operating Point*

Motor performance charts for axis A and B show the worst-case operating point within continuous operation rating. According to SigmaSelect results, the factor of safety in the intermittent range for Axis A is 3.18 and axis B is 3.19.

To better summarize inputs and outputs, the tables below summarize results from SigmaSelect.

*Table 2. Axis B SigmaSelect Summary*

| Rotary Axis, B | | | |
|---|---|---|---|
| *Category* | Parameters | Units | Value |
| *External Forces* | Cutting Force | N | 80 |
| | Applied Torque | Nm | 4 |
| *Geometry* | Table Diameter | m | 0.1 |
| *Calculated Inertia* | Total Inertia at Motor | $kg*m^2$ | **3.45E-06** |
| *Kinetic Profile* | Acceleration | sec | 0.5 |
| | Top Speed | rpm | 100 |
| | Deceleration | sec | 0.5 |
| | Run Time | min | 120 |
| *Gearing* | Gear Efficiency | - | 0.5 |
| | Gear Reduction | - | 30 |
| *Power* | Minimum Power | W | **83.8** |
| *Sigma Select Results* | Required Torque | Nm | 0.272 |
| | Required Speed | rpm | 3000 |
| | Required Power | W | 85.5 |

A SigmaSelect simulation was performed for each axis operating point since inertia and torque values are different. Axis A, the tilt axis, requires twice the torque and significantly more inertia.

*Table 3. Axis A SigmaSelect Summary*

| Tilt Axis, A | | | |
|---|---|---|---|
| *Category* | Parameters | Units | Value |
| *External Forces* | Cutting Force | N | 80 |
| | Applied Torque | Nm | **8** |
| *Geometry* | Part Height | m | 0.1 |
| *Calculated Inertia* | Total Inertia at Motor | kg*m$^2$ | **1.04E-04** |
| *Kinetic Profile* | Acceleration | sec | 0.5 |
| | Top Speed | rpm | 100 |
| | Run Time | min | 120 |
| *Gearing* | Gear Efficiency | - | 0.5 |
| | Gear Reduction | - | 30 |
| *Power* | Minimum Power | W | **167.6** |
| *Sigma Select Results* | Required Torque | Nm | 0.541 |
| | Required Speed | rpm | 3000 |
| | Required Power | W | 170.0 |

It is also important to consider inertial effects on the motor accuracy. As the load inertia increases, the motor has more trouble keeping accuracy when positioning. Each motor is rated for an acceptable inertia, as shown in the Yaskawa Motor Specs table. Though our predicted inertias are below allowable, it is important to will test for inaccuracies with the closed loop system before finalizing the rotary.

*Table 4. Yaskawa Motor Specs*

| | | Motors | |
|---|---|---|---|
| | | SGMJV-01A3A6S | SGMJV-02A3A6S |
| *Torque (Nm)* | Rated | 0.2544 | 0.5096 |
| | Peak | 0.888 | 1.784 |
| *Speed (rpm)* | Rated | 3000 | 3000 |
| | Max | 6000 | 6000 |
| *Inertia ($x10^{-4}kg*m^2$)* | Motor | 0.0665 | 0.259 |
| | Allowable | 1.33 | 3.885 |
| *Body Dim (mm, kg)* | Length | 82.5 | 80 |
| | Width | 40 | 60 |
| | Height | 40 | 60 |
| | Weight | 0.4 | 0.9 |
| *Shaft Dim. (mm)* | Diameter | 8 | 8 |
| | Length | 25 | 25 |
| *Electronics* | Power (W) | 100 | 200 |
| | Voltage (V) | 200 | 200 |
| | Encoder | 20 Bit Absolute | 20 Bit Absolute |

To summarize, the SGMJV 100 and 200 Watt motors are appropriate for the axes B and A, respectively. These motors are conveniently small at only 1-2 lbs and under 3" long.

Despite the extensive design exercise in sizing the right motor, we were given two 200-watt motors without any drawings or motor performance specifications. It is highly recommended not to use these motors until further specifications are obtained for each.

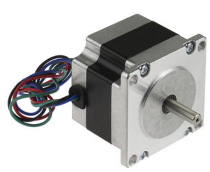


*Figure 19. NEMA 23 Stepper Motor*

Fortunately, we can use our simulation data to source a stepper motor with the appropriate amount of torque. Although stepper motors run much slower than AC servo motors, as long as the torque is sufficient, the rotary will still function. Also, it is important to consider that stepper motors do not have built-in encoders, and will not be as accurate as closed-loop systems.

### 2.4.3 Bearings

Bearings are critical for our design in terms of minimizing run-out and providing the precision we need for our rotary table. Because of this, our rotary table requires bearings specifically designed for machine tools. General practice in the machine tool industry is to use two back-to-back angular contact thrust ball bearings.

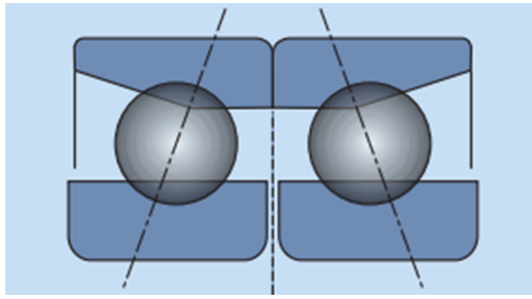


*Figure 20. Back to back bearing configuration*

The preload between angular contact bearings is achieved by clamping a pair of bearings together. Preload increases rigidity, but consequently, it reduces rated speed. Preload also eliminates radial and axial play, increases accuracy and helps to prevent ball skid at high-speed. For our application, at a speed of 100 rpm, a large preload is best. For our spindle the NSK 7006C Series bearings are applicable for our rotary table. They are also easily accessible for ordering.

Assembly using NSK bearings require narrow tolerances for fit, and so to dimension the shaft and housing placement of the bearings, we referred to the manufacturer’s recommendations. The specifications for these bearings are located in Appendix D.

*Figure 21. NSK angular contact bearings*

# 3. Design Development

## 3.1 Requirements/Specifications

After meeting with our sponsors, we determined a list of customer requirements.
The 5th axis rotary table should:

- Work with small Haas CNC machines on light duty machining of wax-type materials at normal production cutting speeds
- Be capable of meeting or exceeding the speeds and accuracies of commercial 5th axis rotaries in the ~$25,000 price range
- Include a way to home the 5th axis mechanism
- Be safe for human operators
- Incorporate appropriately sized servomotors from Yaskawa

In addition to the customer requirements, we created a specifications list to incorporate engineering specifications from which we used to start design process. A reduced list of specifications used for considering preliminary designs is shown below in Table 5. The full specifications list can be seen in Appendix A. The specifications were developed from evaluating constraints of the OM2, comparing specifications from existing products, measured loads, and operating conditions.

*Table 5. Reduced specifications list used for concept generation*

| Number | Feature | Value | Unit | Source | Compliance | Engineering Risk |
|---|---|---|---|---|---|---|
| **1. Geometry** | | | | | | |
| 1.3 | Max Work Table Height | 4 | in | OM2 Specs | Inspection | Medium |
| 1.4 | Max Rotary Height | 10 | in | OM2 Specs | Inspection | Low |
| 1.5B | Work Area Diameter | 3 | in | Rotary Comparison | Inspection | Low |
| 1.6B | Max Work Piece Height | 3 | in | Rotary Comparison | Inspection | Low |
| **2. Kinematics** | | | | | | |
| 2.1 | Rotational Speed | 30 | rpm | Rotary Comparison | Test | High |
| 2.2 | Tilt Speed | 30 | rpm | Rotary Comparison | Test | High |
| **3. Forces** | | | | | | |
| 3.1 | Max Part Weight | 5 | lbf | Rotary Comparison | Test | Low |
| 3.2A | Average Applied Load | 18 | lbf | Spindle Loads | Analysis | Medium |
| 3.3 | Axis A (Tilt) | 6 | ft-lb | Hand Calculation | Analysis | Medium |
| 3.4 | Axis B (Rotary) | 3 | ft-lb | Hand Calculation | Analysis | Medium |
| **4. Energy** | | | | | | |
| 4.1 | AC Motors OM2 Compatible | - | - | Compatible with Haas | Test | Low |
| **5. Materials** | | | | | | |
| 5.3 | Wax machining | - | - | Sponsor Requirement | Test | Medium |
| **10. Assembly** | | | | | | |
| 10.1 | Instructions | - | - | Sponsor Requirement | Inspection | Low |
| 10.2 | Compatible with T-slots | - | - | Sponsor Requirement | Inspection | Low |
| **12. Usage / Operation** | | | | | | |
| 12.1 | Workholder Bolt Pattern | - | - | Rotary Comparison | Inspection | Low |
| 12.3 | Simultaneous machining | 5 | axes | Demand | Sponsor Requirement | Test |

## 3.2 Concept Generation

To begin the ideation process we created a morphological matrix in which solutions were generated for each sub-function of our design. These sub functions or sub-components included the type of transmission for the 4th and 5th axes, how the work piece was to be mounted, how the rotary was to be mounted on the table, the bearings used, and the geometry between the 4th and 5$^{th}$ axes.

*Table 6. Morphological matrix used to group solutions for concept generation*

| Subfunction | Solution | | | | |
|---|---|---|---|---|---|
| Work Part Mounting | Hole Pattern | Collet (Hand tight) | Collet (Automatic) | | |
| 4th Axis Transmission | Belt | Worm | Bevel | Spur | Planetary | Direct Drive |
| 5th Axis Transmission | Belt | Worm | Bevel | Spur | | |
| Table Mounting | Bolt Pattern | Clamps | | | | |
| Bearings | Ball | Tapered Roller | Journal | Thrust | Needle | |
| 4th & 5th axis connection | Inline | Swing Arm | Trunnion | | | |

From the morphological matrix, solutions were combined to create hundreds of complete concepts. In order to produce a more selective list of concepts to choose from, we eliminated some solutions from each sub-function if the solution was not feasible, or other solutions seemed to be much better options in terms of performance, manufacturability, and availability. The options in blue are the solutions we decided to use as concepts. We decided not to include bearings in the detailed drawings because the type will be determined by the rest of the design as well as space constraints. Since a hole pattern and bolt pattern were our only options, the concepts focused on the decision of the transmission for the 4$^{th}$ and 5$^{th}$ axes and the structure between them. From these concepts we chose 11 to draw out in detail, which are listed in Table 7 and attached in Appendix B. Two of the chosen concepts were "wild card" drawings to explore the pros and cons of concepts that satisfy the criteria but have an unexpected design. Each drawing was discussed in terms of how well the design would suit the goal of the project. We evaluated all of the sketches in terms of the criteria in the table below and rated each sketch a value of 1 to 4 where a 1 was given if the design was least desirable in fulfilling the criteria and 4 was the most.

*Table 7. List of chosen concepts from the morphological matrix.*

| Solution | 4$^{th}$ Axis– 5$^{th}$ Axis | Connection |
|---|---|---|
| 1 | Planetary - Belt | Trunnion |
| 2 | Belt – Belt | Inline |
| 3 | Direct Drive - Worm | Swing Arm |
| 4 | DABS | Swing Arm |
| 5 | DASS | Trunnion |
| 6 | Spur - Worm | Inline |
| 7 | Worm - Belt | Trunnion |
| 8 | Worm - Worm | Trunnion |
| 9 | Worm - Worm | Inline |
| 10 | Worm - Worm | Swing Arm |
| 11 | Belt - Worm | Swing Arm |

*Table 8. Decision Matrix evaluating each concept according to important criteria*

| Criteria | Solution # | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| Minimal Workpiece Height | 1 | 1 | 4 | 2 | 4 | 3 | 1 | 4 | 3 | 4 | 4 |
| Minimal Backlash | 1 | 4 | 3 | 4 | 3 | 2 | 3 | 3 | 3 | 3 | 3 |
| High Rigidity | 3 | 1 | 3 | 2 | 3 | 2 | 3 | 4 | 3 | 3 | 2 |
| Minimal Weight | 1 | 4 | 1 | 1 | 2 | 3 | 2 | 2 | 4 | 4 | 4 |
| High Efficiency | 2 | 4 | 2 | 4 | 2 | 1 | 2 | 1 | 1 | 1 | 2 |
| High Manufacturability | 3 | 3 | 4 | 1 | 2 | 3 | 3 | 2 | 3 | 3 | 3 |
| | | | | | | | | | | | |
| **Total** | 11 | 17 | 17 | 14 | 16 | 14 | 14 | 16 | 17 | 18 | 18 |

From the decision matrix, the results showed two solutions that would be the best at meeting the criteria we set for the project. To determine the final design, we looked at the differences between the belt drive and the worm drive option. The belt drive option was eliminated due to torque limitations. The belt drive would also not have a high enough reduction with the motor options that are available to us to achieve the appropriate rpm. In the following chapter we discuss the reasons why we believe the worm option for both axes is the best to pursue for the final design.

## 3.3 Initial Design Considerations

### 3.3.1 Support Design

The design of a modern rotary table is a complex task, typically limited to large corporations with decades of experience in the matter. We will design a single rotary stage that could be manufactured twice and joined to yield a full 5$^{th}$ axis table. The primary engineering features that preceded our design and how they apply to the design of our first prototype follows.

***Structure***

The structure of the machine tool is of primary importance. We do not expect our table to yield, but our design is stiffness limited, with cutting forces causing deflection and making loss of accuracy our primary concern. It is also important to stay above the natural frequency of our applied loads by ensuring the structure is stiff enough, and counteract any harmonics with a damped design.

Two materials most suitable for this application are cast iron and epoxy-granite composite. Cast iron has a yearlong settling time after casting, and as such is unacceptable for our timeline. Epoxy granite is a composite material consisting of granite and quartz particles in an epoxy matrix, and has one of the highest damping factors for any material. Its downside is its low strength, which we have considered improving by coming up with a new epoxy-granite-fiber matrix wherein we add carbon or glass chopped fibers to the mix to improve stiffness and strength. This will be investigated in the future, due to the large R&D required to manufacture such a material. We are designing around 6061-T6 AL due to availability and comparable stiffness to epoxy granite.

The geometry of our housing was determined to be double-cantilever as shown in Figure 22. This is beneficial because it allows us to design one rotary for both the A and B axes.

The major elements of the housing are that it must hold tapered roller bearings, a preload mechanism, a worm drive, and a circular bore for a platter. It is a uni-body design, with the entire structure to be machined out of aluminum.

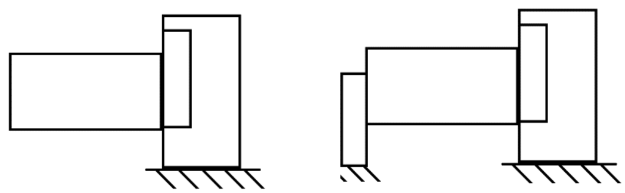


*Figure 22. Geometry Configuration, Cantilever-Cantilever and Cantilever-Fixed*

### Reliability

A major limitation in our design is gear lifetime. Preloaded and low backlash gear designs typically experience higher wear than conventional gear trains. Assuming a 20 hour workweek in a light setting, with a 3 year lifetime, our design requirement is 3000 hours or operating time. This may not be possible without expensive gears, and as such we are open to making regular gear or worm replacement mandatory. If this is the case, the replacement of such parts must be easy for an average technician.

### Anti-Backlash Design

We evaluated anti-backlash methods on a numerical basis, where a score of 1 is the lowest score and 4 is the highest score. Table 9 shows our decision matrix results.

*Table 9. Decision Matrix for Backlash Reduction Methods*

| | | Solutions | | | | | |
|---|---|---|---|---|---|---|---|
| | | Precision Ground Gears | Cone/Hourglass Gears | Duplex Worm | Anti-Backlash Gears | Preloaded Mesh | Spring Loaded Mesh |
| Criteria | High Manufacturability | 1 | 1 | 2 | 3 | 4 | 4 |
| | High Ease of Alignment | 1 | 1 | 1 | 3 | 4 | 4 |
| | Minimal Space Constraint | 4 | 4 | 4 | 3 | 2 | 2 |
| | Maximum Longevity | 4 | 4 | 2 | 2 | 2 | 1 |
| | Minimum Maintenance | 3 | 3 | 2 | 2 | 1 | 4 |
| | High Off-The-Shelf Components | 1 | 1 | 1 | 3 | 4 | 4 |
| | Minimum Backlash | 3 | 4 | 3 | 4 | 2 | 4 |
| | **Total** | 17 | 18 | 15 | 20 | 19 | 23 |

According to our decision matrix, the spring loaded mesh is the best design for our application. Although the spring loaded mesh is not expected to last as long as other options, we will test the longevity with a prototype housing and gear mesh. Our plan is to run the setup for 100 hours and test the wear on the worm and gear to predict the lifetime of the setup. If we find the wear to be excessive, our next best option is anti-backlash gears.

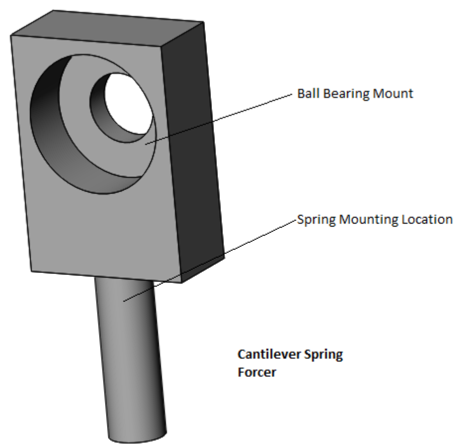



*Figure 23. Cantilever spring forcer*

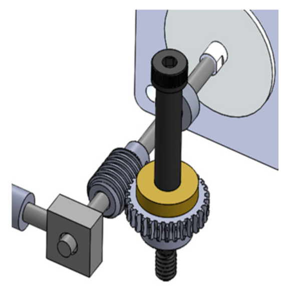


*Figure 24. Worm gear backlash reduction system*

Figures 23 and 24 show our initial design concept for a spring-loaded mesh between the worm and gear. The worm contacts the gear through shaft in bending to eliminate backlash. This design can be made with a spring preloaded on the end of the shaft and the motor fixed to the housing. Additionally, the contact between the worm and the gear can be submerged in oil to decrease friction and increase gear longevity.

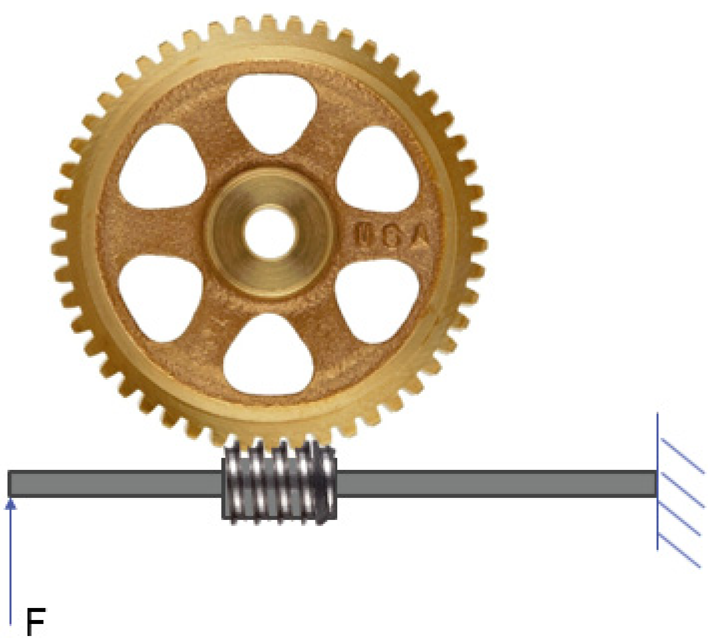



*Figure 25. Method to reduce backlash in worm drive*

### Bearings

The bearings in our design are critical given the high loads that we expect to experience. This problem is further complicated by the fact that cost is a major issue. We designed our preliminary rotary housing to use Timken tapered roller bearings. These SET45 Timken bearings are readily available at auto parts stores, and the runout on several can be inspected before purchase, allowing us to use low runout bearing through the laws of averages.

*Figure 26. Tapered Roller Bearings*

*Figure 27. Thrust bearing*

Next, 24mm diameter thrust bearings are designed to be used in conjuction with the tapered roller bearings to provied the necessary 1500 lbf of preload when torqued properly with a Grade 2 ¼-20 thread.

### Prototype 1 Design

The end result of this is a rotary platform that is both low cost and highly accurate. For the PDR we purchased a NEMA 23 stepper and motor controller to evaluate the geartrain efficiency and run tests before comitting to a servo from Yaskawa. This is because friction factors are hard to predict, and empirical data is always more accurate. After we complete a functioning prototype, we will be able to iterate our design and properly pick a motor. Figure 28 shows images of our rotary table design.

We manufactured a single rotary as described above and have completed testing on it. Both during the manufacturing process and during testing we determined some design changes that

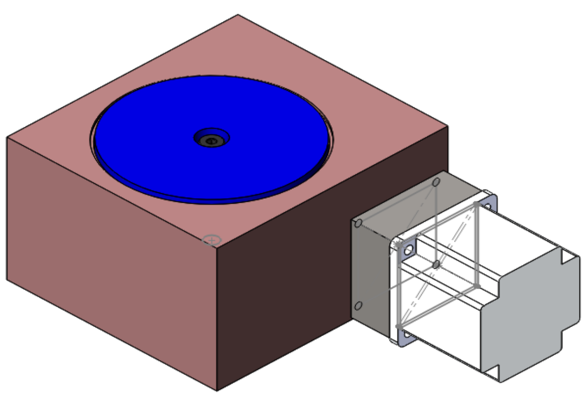


*Figure 28. Gen5 rotary table with motor placement*

would need to be made for this rotary to be completely functional. We had noticed that after manipulating the rotary that the worm wheel was not meshing with the worm gear. The shoulder bolt used for shaft was not rigid enough to maintain straightness and runout. Many of the dimensions and fits needed to be reconsidered for assembly as well. The cavity for the gear was inaccessable for assembly and the gear itself appeared to be too small to move the rotary. We used this knowledge to create a design for our second prototype, which we will be described on the following section.

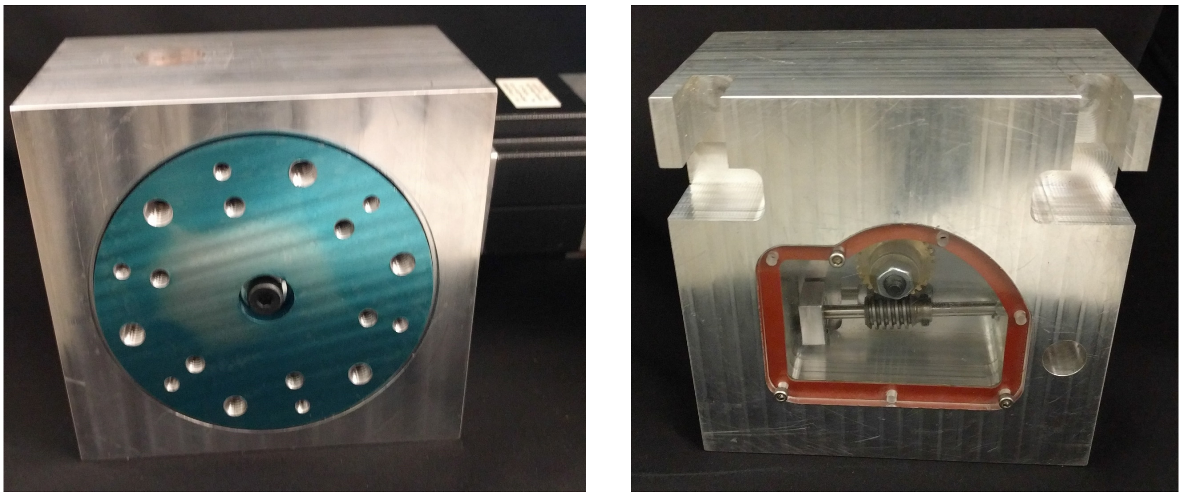


*Figure 29. Completed prototype 1*

# 4. Final Design

## 4.1 Overall Layout

Our final design takes into account lessons learned from our first prototype and in depth calculations. We have chosen two angular contact bearings to hold our spindle and a more robust shaft that is bolted onto the bottom of our platter. We have improved the rigidity of our gear mesh preload design using two dowel pins and a spring plunger. Furthermore, we have changed to larger 16DP gears to give a better factor of safety.

## 4.2 Detailed Design

Individual part drawings can be found in Appendix B for manufactured parts and Appendix D for off the shelf parts.

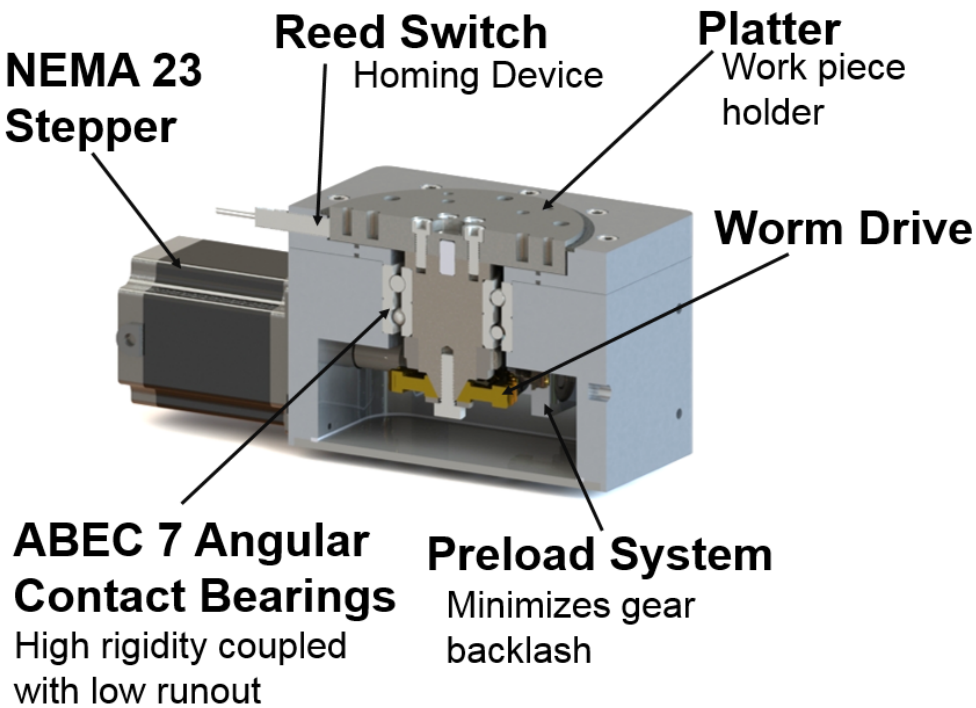



*Figure 30. Rotary Schematic*

### 4.2.1 Cap

The cap holds the outer race of bearing in place and provides the preload in conjunction with the bearing locknut. It also has grooves for O-rings to prevent contaminants from entering the rotary. Half of the reed switch used for homing is contained in the cap. The cap was manufactured using a Haas CNC Mill.

### 4.2.2 Housing

The housing is to be manufactured from aluminum. It was redesigned to accommodate our new internal configuration. There are different configurations for the B axis and the A axis. The features on both stages are nearly identical with the only difference being that the A axis has more material on the bottom of the stage to lift the B axis higher to prevent it from crashing into the table. The housing was manufactured using a Haas CNC Mill.

### 4.2.3 Platter

The updated design consists of a detachable platter that is bolted to the shaft. The platter has the same hole pattern as the TRT-100. It will be made out of steel, with a reed switch located on the side for homing. The platter also has a bore in order to accommodate fixtures. The platter was machined on a Haas CNC lathe.

### 4.2.4 Shaft

The shaft is precision manufactured of a Haas CNC lathe to accommodate the tight tolerances required to shrink fit the NSK bearings. Below the bearings the shaft is threaded for a bearing locknut to provide preload to the bearings. Lastly, a conical section locates and fastens the gear to the shaft. The shaft will also be manufactured on a Haas CNC lathe.

### 4.2.5 Gear Preload

In order to prevent deflection, two dowel pins hold the gear preload mechanism while a spring plunger provides the preload force. The spring plunger will have to be adjusted over time as the

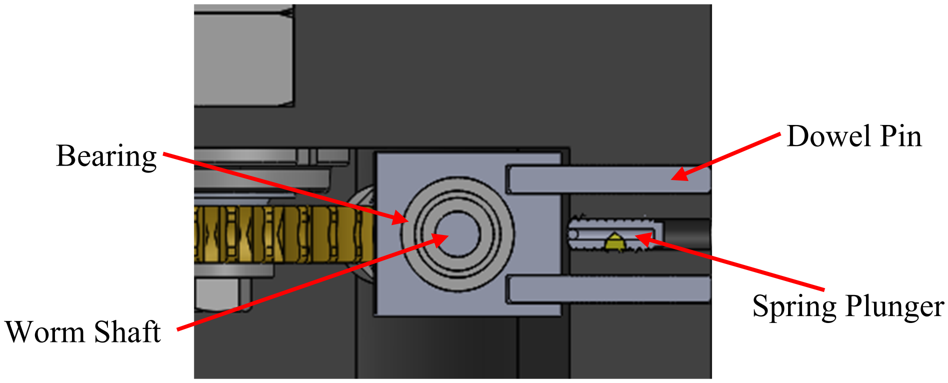



*Figure 31. Gear preload system*

gears will wear out, which just requires a simple turn with a screwdriver.

### 4.2.6 Worm Wheel

The worm wheel is an off-the-shelf part that was modified for our rotary. A conical countersink is machined into the worm wheel to perfectly contact the spindle and for proper concentric alignment. This design allows the operator to easily replace the worm wheel for another when worn out.

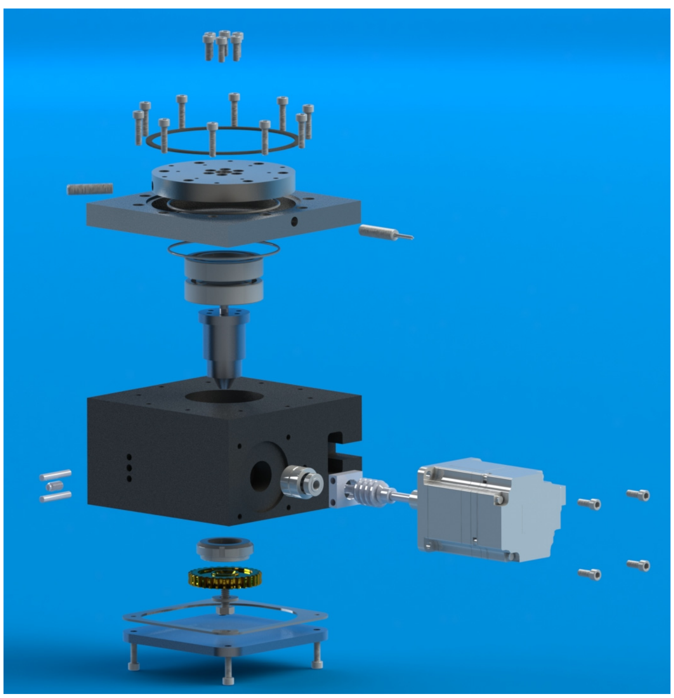


*Figure 32. Rendering of rotary exploded view*

## 4.3 Analysis Results

### 4.3.1 Drive Shaft

The drive shaft is connected directly to the motor output shaft and transmits power to platter. Stresses in the drive shaft include bending, torsion, and shear and must also be rated for endurance limit. The drive shaft code solves for the necessary spring constant, deflection, and preload values. Furthermore, the solution verifies endurance limit fatigue strength with the parameters calculated previously. Results from the drive shaft calculation show a 2.98 factor of safety against fatigue failure.

### 4.3.2 Driven Shaft

The driven shaft refers to the spindle shaft attached directly to the gear. Since the spindle is tapered at the contact point with the gear, the shaft is analyzed at its smallest diameter, determined by the gear through hole. Loads for this shaft are primarily torsion, and is tested for endurance strength with Marin Factors. In conclusion, the driven shaft is rated well above endurance fatigue with a 20.67 factor of safety.

### 4.3.3 Inertia

Each Yaskawa AC servo motor is specified with an allowable inertia value that will allow for acceptable motor response. If inertia values exceed those specified by the manufacturer, the motor will not be accurate in positioning the rotary. This code calculates the inertia due to the platter, motor, housing, drive shaft, worm gear, and coupling as seen by each axis motor separately. Inertia values for each axis is relatively low due to the 30:1 reflected inertia ratio and give a factor of safety of 3.73 against response error.

### 4.3.4 Worm Gear A and B

This code evaluates the worm gear for 25000 hours of life using the AGMA method. It is assumed that the worm will outlast the gear by a large margin due to material properties. In an effort to share parts between the rotaries, each axis will use a 16 diametral pitch worm and gear. As these gears are critical to the function of our design, the factor of safety against fatigue failure over 25000 hours is 2.80 and 5.04 for axes A and B respectively.

### 4.3.5 Bearings

It is important that our bearings are rated to the expected loads on the rotary. The most critical parameter is the spacing between the bearings and their ability hold an applied moment. To save space in the rotary, two NSK precision bearings are preloaded without a spacer. Without a spacer, the bearings are still capable of the expected loads with a minimum factor of safety of 6.6.

### 4.3.6 Bolts

The code analyzes the bolts that attaches axis B to axis A. There are two sets of bolts to evaluate: the bolts that attach axis B to the fixture plate, and the bolts that attach the fixture plate and axis B to axis A. The largest stress the bolts will see is shear stress. ¼”-20 Grade 1 Bolts were used for the analysis, resulting in a safety factor of 90.63 for the bolts that attach to axis B and a safety factor of 45.32 for the bolts that attach to axis A. The minimum length of engagement for the threads of the bolts is 0.2856 inches.

### 4.3.7 Summary

The table below summarizes the calculation results with factor of safety and calculation method from Shigley's Engineering Design.

*Table 10. Factors of Safety for Components*

| Part | Longevity | Factor of Safety | Reference |
|---|---|---|---|
| **Axis B Gear** | 25000 hrs | 5.04 | AGMA Method |
| **Axis A Gear** | 25000 hrs | 2.80 | AGMA Method |
| **Drive Shaft** | Endurance Limit | 2.98 | Marin Factors |
| **Driven Shaft** | Endurance Limit | 20.67 | Marin Factors |
| **Axis A Bolt Fixture** | Static Loading | 45.32 | Shear |
| **Axis B Bolt Fixture** | Static Loading | 90.63 | Shear |
| **Axis B Bearings** | Static Loading | 15.4 | Bearing Load Rating |
| **Axis B Bearings** | Static Loading | 85.56 | Axial Rating |
| **Axis A Bearings** | Static Loading | 6.6 | Bearing Load Rating |
| **Axis A Bearings** | Static Loading | 40.53 | Axial Rating |

## 4.4 Cost Analysis

We have analyzed the cost of both rotaries in a worst case scenario, as shown in the table below. The cost of the 5$^{th}$ axis that we are building for our sponsor will be much lower, as we are getting the motors donated from Yaskawa and the bearings at a lower price on eBay. If this were to be manufactured by someone else, they could research other sources to buy quality parts at a reduced cost. If this were to be mass produced, costs would also be reduced by purchasing in mass quantities.

*Table 11. Complete list of parts and cost for both rotaries*

| Cost Analysis | | | | |
|---|---|---|---|---|
| **Part/Material** | **Qty** | **Supplier** | **Supplier #** | **Cost** |
| Steel | 1 | IME Dept | - | N/A |
| Aluminum | 1 | McMaster | | 290.00 |
| 0.25"x0.5" Dowel Pins | 6 | McMaster | 97395A475 | 10.21 |
| 3/16"x 1.25" Dowel Pins | 4 | McMaster | 97395474 | 11.25 |
| Ball-Nose Spring Plunger | 2 | McMaster | 3408A75 | 7.24 |
| 10-32 x 0.5" Socket Head Cap Screws | 13 | McMaster | 92185A989 | 4.26 |
| 10-32 x 0.75" Socket Head Cap Screws | 24 | McMaster | 96006A693 | 5.71 |
| Gasket Material - Neoprene | 2 | McMaster | 8837K112 | 21.70 |
| 75mm x 3mm wide O-ring | 4 | McMaster | 9262K409 | 6.26 |
| 1-9/16"Bearing Lock Nut | 2 | McMaster | 6343K160 | 14.44 |
| 3mm Set Screw | 2 | McMaster | 92015A101 | 9.34 |
| 1/4" Shaft | 2 | McMaster | 1144K11 | 24.15 |
| 1/4" ID Washer | 2 | McMaster | 98019A360 | 6.75 |
| 3/4"x.75 Cap Screw | 2 | McMaster | 90201A111 | 9.62 |
| Small Ball Bearing | 2 | McMaster | 57155K388 | 9.56 |
| 14mm Ball bearing | 2 | McMaster | 2423K24 | 38.22 |
| 1/4"-20 Socket Head bolts | 8 | McMaster | 90128A242 | 9.09 |
| 1/4"-8mm Shaft Coupling | 1 | McMaster | 2764K123 | 60.54 |
| 1/4"-14mm Shaft Coupling | 1 | McMaster | 2764K322 | 78.69 |
| 7006C NSK Bearings | 4 | McMaster | 2385K46 | 1,091.00 |
| SPDT 150V DC Reed Switch | 2 | McMaster | 6585K22 | 92.22 |
| 30T 16DP Gear | 2 | Boston Gear | G1043 | 91.60 |
| 16DP Worm | 2 | Boston Gear | LVHB1 | 46.40 |
| 200W Motor + Cable | 1 | Yaskawa | SGMJV-02A | 550.00 |
| 100W Motor + Cable | 1 | Yaskawa | SGMJV-01A | 500.00 |
| | | | | |
| **Other Items** | | | | |
| SAE 80 Oil | 1 | McMaster | 3512K63 | 10.00 |
| Emery Cloth | 1 | McMaster | 8238A53 | 9.93 |
| Cylinder Hone | 1 | Ebay | - | 20.00 |
| Spanner Wrench | 1 | McMaster | 6975A16 | 19.58 |
| | | | Total | 3,047.76 |

# 5. Product Realization

## 5.1 Manufacturing

The nature of the rotary table requires a very high level of precision, typically beyond the levels we would be able to achieve in a school setting without precision scraping. This is due to the fact that our system design features a long tolerance chain from the machine table to the rotating B axis. This chain, as it were, is composed of a number of different machined or ground components which must be maintained within very tight tolerances because their affect on system accuracy is compounded greatly throughout the project.

*Table 12. Stackup Chain*

| System Tolerance Stackup Chain | | | |
|---|---|---|---|
| Tolerance Feature | Moment Arm | Material | Tolerance |
| Housing A Perpendicularity of bottom surface to bearing bore | 175mm | 6061AL | 2 microns |
| Bearing set A outer race to inner race parallelism | 27.5mm | Tool Steel | 2.5 microns |
| Spindle A perpendicularity of datum surface to center axis | 10mm | Steel | 5 microns |
| Platter A parallelism from top to bottom plane | 12.6mm | Steel | 2.5 microns |
| Adaptor plate perpendicularity from A to B mounting plane | 100mm | 6061AL | 20 microns |
| Housing B Perpendicularity of bottom surface to bearing bore | 100mm | 6061AL | 2 microns |
| Bearing set B outer race to inner race parallelism | 27.5mm | Tool Steel | 2.5 microns |
| Spindle B perpendicularity of datum surface to center axis | 10mm | Steel | 5 microns |
| Platter B parallelism from top to bottom plane | 12.6mm | Steel | 2.5 microns |
| Summation | | | **44 microns** |

The above table illustrates the fact that even with incredible precision beyond 99 percent of most machine shops in the world, the total tolerance stackup is over 44 microns, which frankly is unacceptable for a rotary table. In order to remedy this situation, we used unique machining techniques to achieve high levels of precision in only our critical dimensions, and individually matched parts to hit our tolerances. This was time consuming and took hundreds of hours.

*Figure 33. Housing Maching Set Up*

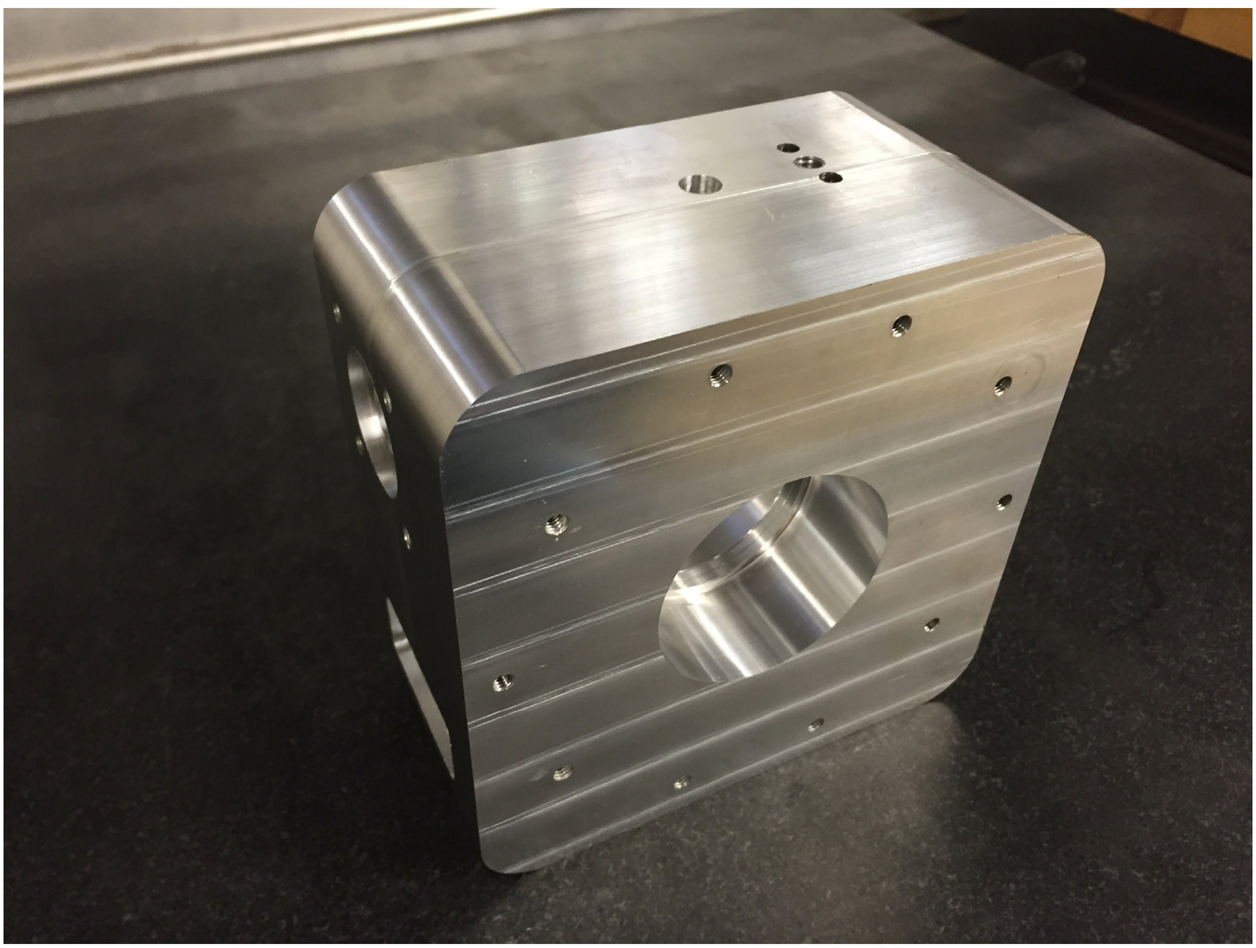


*Figure 34. Housing inspection*

The most critical surfaces we machined were arguably the housing. It is an incredibly complex system and arguably making one of these would be worthy of a senior project in itself. The precision, fit and finish of the component are astounding. To maintain perpendicularity of the bore to the bottom surface, both of these were machined in the same machine tool, using the same endmill, in the same operation, with no less than 5 unique finishing passes.

In order to maintain bore toleraces we had to push the ballscrew accuracy of the VF3 to its limits. We used in machine probing in concert with in machine gauging to precisely control our bore size, which had a tolerance of plus or minus 2 microns. The process involve finish machining the bore twice, probing the size and comparing it to the expected size, then probing a gauge block and finding the probe inaccuracy, and using this to compensate for future probing. Then the bore was finish machined again, with a 25 micron chipload, the bore was then probed to find the actual chipload, and this offset was used to make the final finishing pass in the machine.

*Figure 35. Spindle Machining*

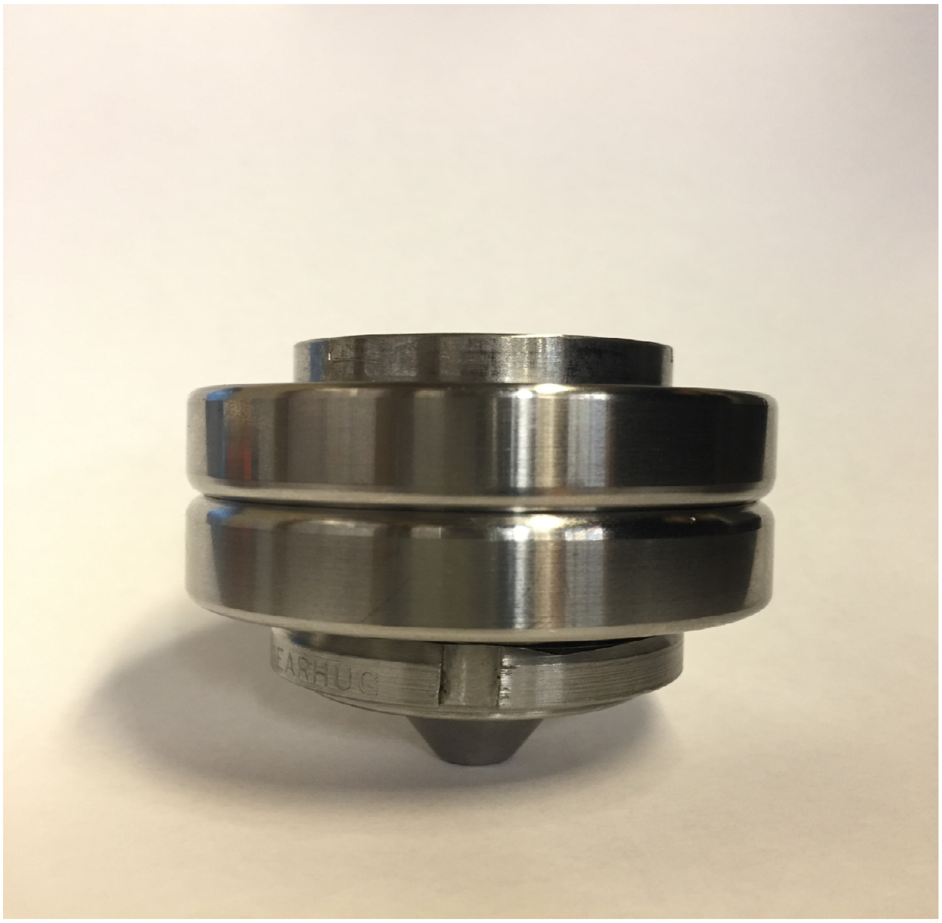


*Figure 36. Spindle Assembly*

The spindly was also a critical assembly. The central spindle was machined entire in one operation, and then to hold the plus or minus tolerance of 2.5 microns on the inner bearing race mounting feature, the OD was lightly buffed to size. Following this 20 micron shims were used to ensure the length was accurate to micron levels, this allows us to preload the NSK bearings with the shown spindle nut. The whole assembly drops into the housing with only a 20-degree temperature differential thanks to the tight tolerances. Following this the top surface of the spindle was surface ground parallel to the housing bottom surface.

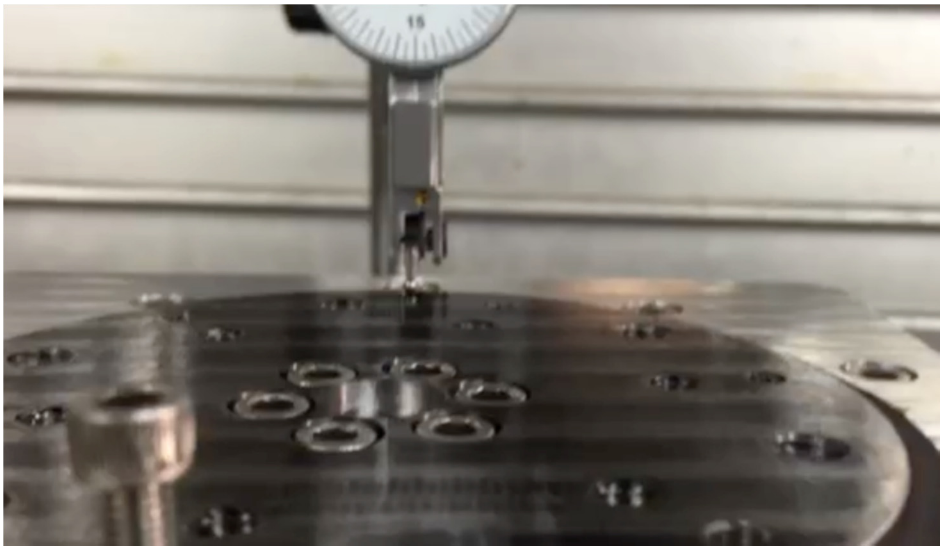



*Figure 37. Perpendicularity Measurement*

Ultimately the difficulties we faced in making such a complicated system were overcome, but the workload involved was considerable to say the least. This is to be accepted, there is no way around tight tolerances with such a system, and we were successful in our manufacturing endeavors.

## 5.2 Design Edits

Our planned design described previously in this report was to use Yaskawa servomotors in conjunction with the 4th and 5th axis drive cards from Haas, both of which were donated for our project. In order for the motor to interface with the $4^{th}$ and $5^{th}$ axis driver cards, we were sent the exact servo motors that Haas uses on their rotaries. Because of this, the motors were much larger and heavier than what we had been planned for and so they could not be used with our design. This was determined very late in our project, so our sponsor requested that we manufacture just a fourth axis that is compatible with Yaskawa's donated motor.

In order to complete the project that our sponsors originally asked for, we decided to create a complete 5th axis that was run with stepper motors and an external controller. A G-code macro was written that calculates the position of each axis and outputs them through the serial line as text. This text line is sent to the microprocessor of our controller, an Arduino Uno, which parses the string into recognizable commands and variables. If the letter H is sent, each axis is homed, by spinning until the reed switch registers and then moving to a predetermined offset. Otherwise, each axis can be sent a direction and number of degrees and the Arduino will calculate how many steps to turn the stepper to move to the new angle.

Pictured below is the controller, which contains two AC to DC power supplies, a 24 Volt one for the motors, and a 12V one to power the Arduino. Pololu DRV8825 stepper drivers are used to send commands to the steppers. A MAX3223 breakout from Sparkfun is used to convert the RS232 input to TTL so that the Arduino can properly receive the text. We chose to use NEMA 23 stepper motors with 179 oz. in. of torque in order to withstand machining loads and maneuver the weight of the B-axis.

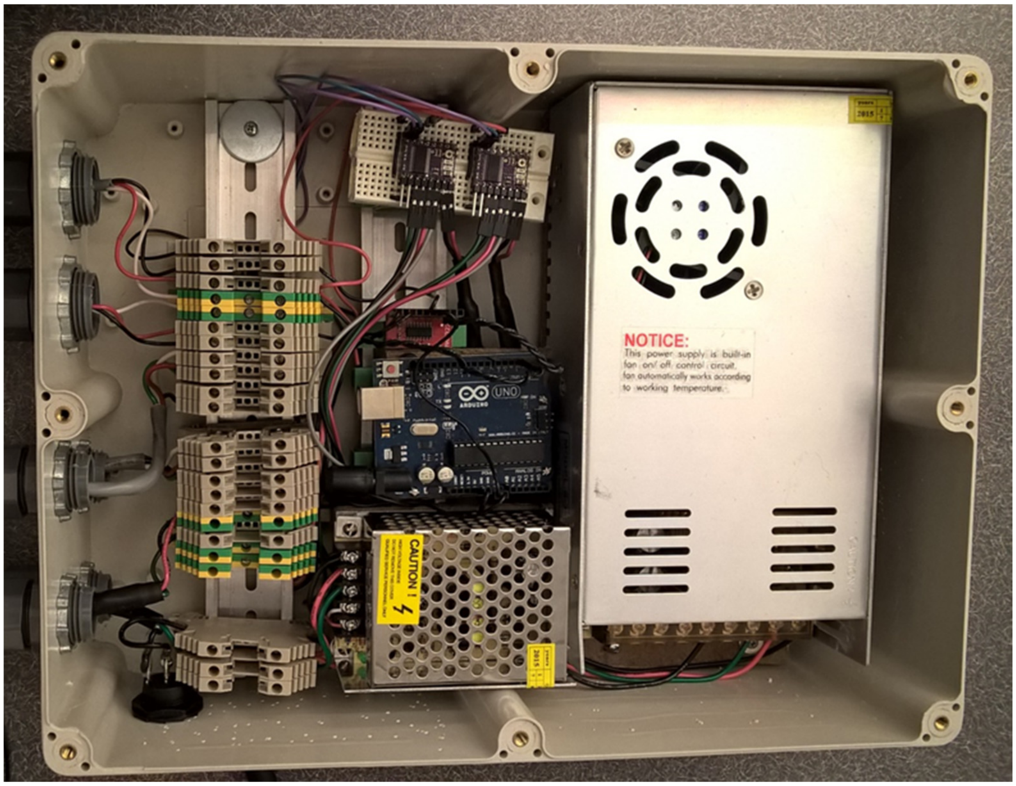



*Figure 38. Custom stepper motor controller*

## 5.3 Recommendations for Future Manufacturing

While we have gone through three prototypes during our senior project in order to refine the design and improve manufacturing, there are still a few changes to make in terms of machining these parts. Dimensions that have fits should be machined to nominal sizing, and then reamed for fit. Most features should be designed for a slip fit rather than an interference fit if the mating is not crucial, as it will eliminate the need to risk the part during pressing. If the feature is crucial, machine undersize, and check the dimension before re-machining to size. The accuracy of our CNC machines were not up to par for machining tolerances of fixtures and so the dimensions had to be manually accounted for by the operator. Dimensions should be checked after machining each part and before assembly. Also, it is crucial to consider flatness tolerances on the platter of the rotary, and so the spindle and the platter should be surface ground before assembly.

# 6. Design Verification Plan

## 6.1 DVP&R

Documentation for our design verification plan and report can be found in Appendix F. Below are testing procedures for all of the planned tests.

## 6.2 Vibration Testing

Background

Vibration testing uses a sine sweep to find the natural frequency of the rotary. Operating at the natural frequency of the rotary can cause inaccuracies in machining, and possible structural failure.

**Equipment**

- Rotary
- Accelerometer
- Signal Analyzer
- Blue Box Charge Amplifier
- Shake Table

**Procedure**

1. Secure the rotary table to the shake table and attach the accelerometer with a dab of wax, as seen in the set-up figure.
2. Connect the charge amplifier to the accelerometer and channel two of the signal analyzer.
3. Preset the signal analyzer to sine sweep between 10-2000 Hz, where the input is channel one and output is channel two. Display the Bode Diagram with a linear magnitude.
4. Connect the signal analyzer source to channel one and the input for the shake table controller via a tee junction.
5. Turn on the hydraulic pump and follow directions posted on the shake table control panel.
6. Press “start” on the signal analyzer to start the sine sweep and record data.
7. Trace results to find each mode of natural frequency and print the bode diagram.
8. Repeat the procedure at least three times with the housing assembled and disassembled.

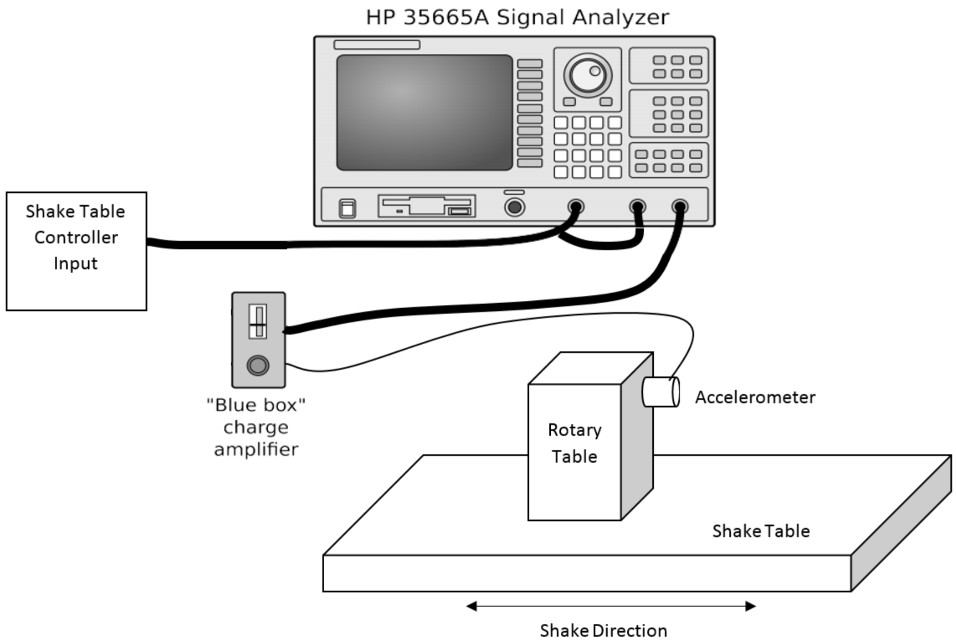



*Figure 39. Laboratory setup diagram*

## Setup

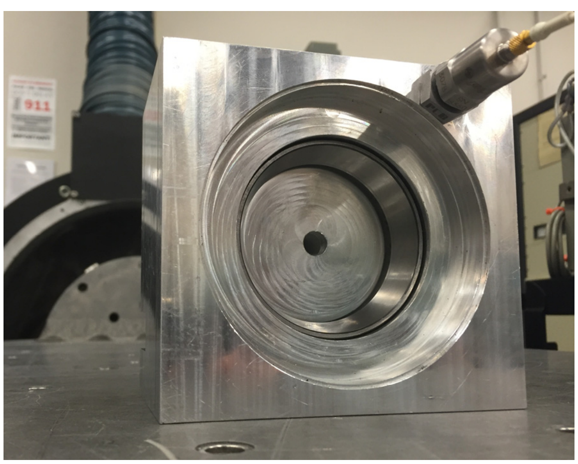


*Figure 40. Accelerometer mounting configuration on housing with components removed*

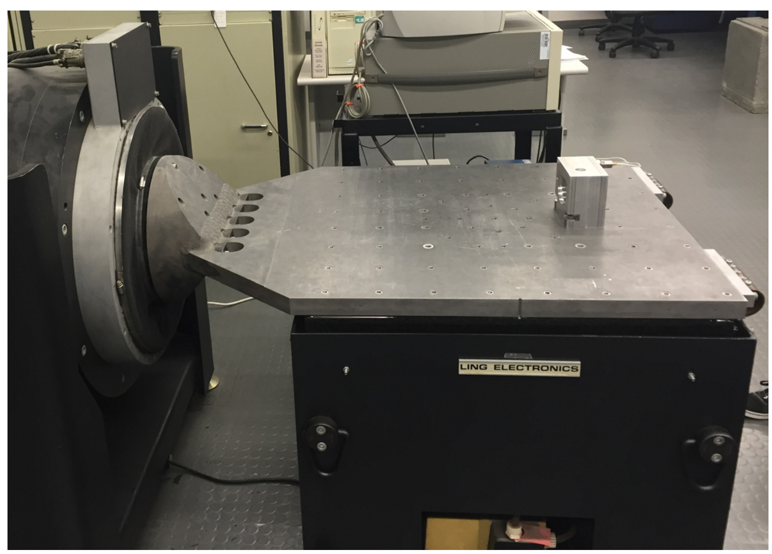



*Figure 41. Housing mounted to the shake table*

**Results**

This testing was completed on February 12th, 2016 with prototype 1 in the Mechanical Engineering Vibrations Laboratory. The results showed a first modal natural frequency of 380Hz and a second mode at 1200Hz. The sweep covered 10Hz to 2000Hz and was repeated 9 times with different housing configurations. The graph of the test completed with all of the parts removed from the housing can be seen below. This testing shows that aluminum may not be a viable material for our housing since the first two modal frequencies are both at values that will be reached by the spindle.

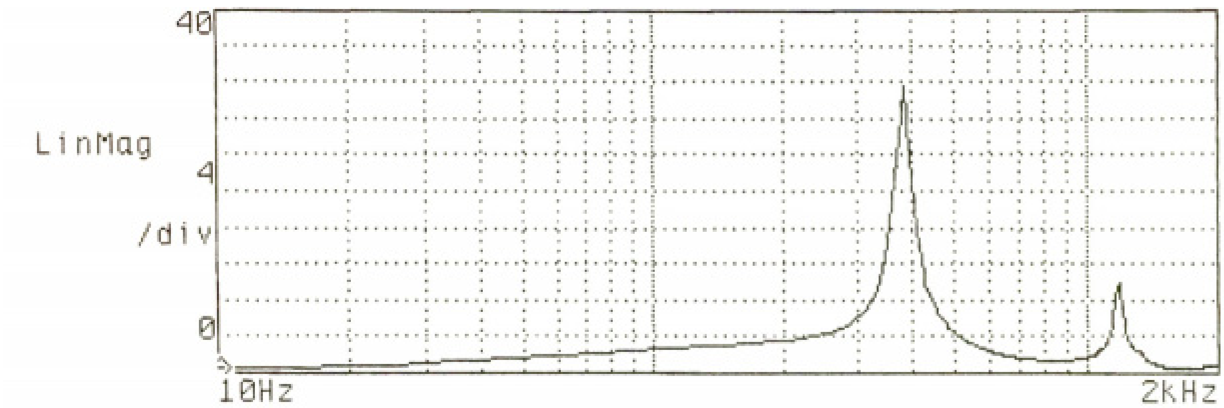



*Figure 42. Results of vibration testing completed on the housing alone.*

## 6.3 Backlash Testing

**Equipment**

- Assembled Stage
- Screwdriver

**Procedure**

1. Place assembled housing on its side with the spring plunger facing up on the housing.
2. Tighten the spring plunger towards the gear.
3. Rotate the platter to feel the backlash.
4. Continue until no noticeable play could be detected.

## 6.3 Tolerance Testing

Background

We must check finished manufactured parts with the tolerances specified on drawings.

Dimensions Check

**Equipment**

- Dial Caliper
- Micrometer set (0 – 4”)
- Bore gauges
- Optical Comparator
- Housing

**Procedure**

1. Measure all linear dimensions as mentioned in manufacturing drawings with a caliper.
2. Measure all external diameters as mentioned in manufacturing drawings with the appropriate sized micrometer. (0” – 1”, 1” – 2”, etc.)
3. Measure all internal diameters as mentioned in manufacturing drawings by extending the appropriately sized bore gauge, and measuring the ends of the bore gauge with a micrometer.
4. Measure difficult to reach internal dimensions using the optical comparator.

Flatness

**Equipment**

- Magnetic Base
- Granite Table
- Dial indicator
- Housing

**Procedure**

1. Place Housing on a micro-flat granite table.
2. Attach dial indicator to magnetic base and lower the dial indicator onto part until the tip is depressed.
3. Run dial indicator across the part and measure the height at 5 points along the surface.
4. Find the range of points across the surface. The value is the flatness tolerance of the part.
5. Using datums noted on drawings at MMC, see if dimensions are within tolerance.

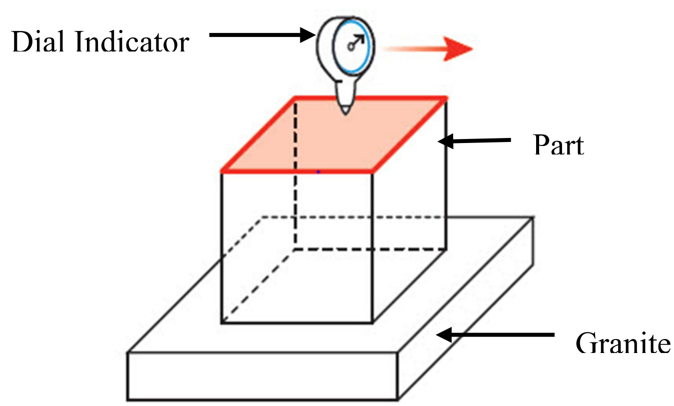



*Figure 43. Setup to measure flatness*

Run-out

**Equipment**

- Magnetic Base
- Dial Indicator
- Assembled Housing

**Procedure**

1. Place each stage on a flat surface. Attach the dial indicator to the magnetic base and place on flat surface.
2. Have the test dial indicator touching the part.
3. Rotate the platter while moving the indicator along the part.
4. Measure the total variation in run-out.
5. Adjust part to best eliminate run-out.

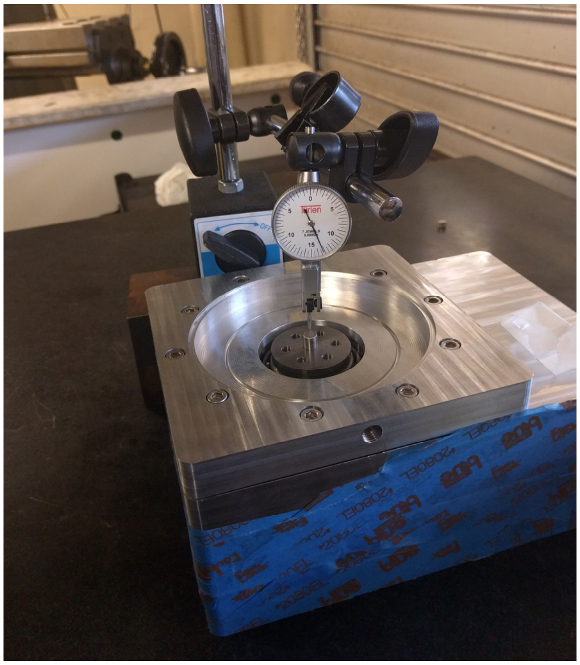


*Figure 44. Setup for measuring run-out tolerance*

**Results**

The majority of our parts were within the tolerance specified for each part. Some parts had to go back to the machining process to guarantee that dimensions were met. The overall run-out on the platter of each housing was about Ø 0.001”.

## 6.4 Efficiency Testing

**Equipment**

- Yaskawa Servo Motor
- Controller
- Data Acquisition
- Rotary

**Procedure**

1. Measure the moment of inertia for each part in motion.
2. Pick an angular acceleration within capability of the servo motor.
3. Calculate the torque required to accelerate the calculated moment of inertia. This will require adding torques from objects spinning at different speeds.

   Where, $T\ [N * m] = I\ [kg * m^2] *\ \alpha\ [\frac{rad}{sec^2}]$
4. Write a program that starts the motor at a constant angular speed and accelerates at the rate above. Do not measure data if motor is standstill to avoid error with inrush current.
5. Perform the acceleration with the Yaskawa motor and record torque results.
6. Compare the torque measured at the motor with the torque calculated.
7. Repeat several times at different acceleration values.

**Results**

We were not able to perform this test because the choice of motor changed many times during the last stages of build process due to certain circumstances. If there was more time allotted, we would be able to perform this test.

## 6.5 Gear Wear Testing

<u>Background</u>

As the rotary runs, the gears will begin to wear, so this test gives a timeline of the gear functionality.

**Equipment**

- Housing (worm and gear)
- Stepper motor
- Arduino motor controller
- Power source
- 2 C-Clamps

**Procedure**

1. Measure diameter of gear and worm.
2. Clamp housing onto a flat surface.
3. Put dial indicator on the magnetic base and secure to a metal surface.
4. Connect Arduino to power source.
5. Connect motor to the Arduino controller.
6. Run a random angle program.

7. Turn off power and disconnect everything.
8. Measure diameter of gear and worm and record the values along with how long the system was running

**Results**

This test was unable to be conducted because the amount of time required to run this test was not available to us at the time the rotaries were completed, especially if the longevity of the gear was predicted to be 250000 hours.

# 7. Conclusions and Recommendations

We have created a robust design to meet the specifications of our sponsors. Our factors of safety ensure a safe mechanism to last our rated 2400 working hours. The majority of our parts are off the shelf from McMaster Carr, and assembly can performed in any basic shop. There are 7 parts to be manufactured in house for each rotary. This allows our product to be created by other schools or home machinists with access to a mill, lathe, and laser cutter. We are confident in our design and in the ability to manufacture and test it.

In terms of improvements that could be made, the rotary could be better designed for assembly. Press fits should be minimized and slip fits should be chosen instead. The preload applied to the bearings were difficult to determine and so a better method needs to be devised in order to provide the appropriate preload. The design could also be more modular, so that subassemblies could be made ahead of time and then assembled together for better efficiency. Also, critical dimensions should be more carefully watched during manufacturing.

There was only limited time to test the motors and the controller, and so it would be recommended to do more extensive testing to understand the system of our rotary table. Originally, we were asked to use the Yaskawa/Haas motors for our rotary table, but it would be best to use non-proprietary motors and software for troubleshooting purposes.

Based off the testing we have completed so far, we would recommend looking into getting the parts in cast iron to negate vibration concerns and lower machining costs. We would also recommend exploring what it would take to modify our design to be able to machine aluminum along with wax. It would also be beneficial to create a document that has step by step instructions to manufacture this rotary.

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# Appendices

# Appendix A

Specifications List

| Specification Sheet | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Project 21. 4th and 5th Rotary Stages | | | | | | | | |
| **Number** | **Feature** | **Value** | **Unit** | **Demand vs. Desire** | **Source** | **Compliance** | **Engineering Risk** | **Notes** |
| 1. Geometry | | | | | | | | |
| 1.1 | Max Rotary Length | 16 | in | Demand | OM2 Specs | Inspection | Low | May interfere with tool change |
| 1.2 | Max Rotary Width | 10 | in | Demand | OM2 Specs | Inspection | Low | |
| 1.3 | Max Work Table Height | 4 | in | Demand | OM2 Specs | Inspection | Medium | |
| 1.4 | Max Rotary Height | 10 | in | Demand | OM2 Specs | Inspection | Low | Cannot interfere with tool change |
| 1.5A | Work Area Diameter | 5 | in | Desire | Rotary Comparison | Inspection | High | Max part : 5" Cylinder |
| 1.5B | Work Area Diameter | 3 | in | Demand | Rotary Comparison | Inspection | Low | Min part : 3" Cylinder |
| 1.6A | Max Work Piece Height | 5 | in | Desire | Rotary Comparison | Inspection | High | Max part : 5" Cylinder |
| 1.6B | Max Work Piece Height | 3 | in | Demand | Rotary Comparison | Inspection | Low | Min part : 3" Cylinder |
| 2. Kinematics | | | | | | | | |
| 2.1 | Rotational Speed | 30 | rpm | Desire | Rotary Comparison | Test | High | TRT100 goes 167rpm |
| 2.2 | Tilt Speed | 30 | rpm | Desire | Rotary Comparison | Test | High | |
| 2.3 | Resolution | 0.001 | deg | Desire | Rotary Comparison | Test | High | Same as the TRT100 |
| 2.4 | Axix B (Rotary) Travel | 360 | deg | Desire | Rotary Comparison | Test | Low | Continuous rotation. Same as the TRT100 |
| 2.5 | Axis A (Tilt) Travel | ±120 | deg | Desire | Rotary Comparison | Test | Low | Same as the TRT100 |
| 3. Forces | | | | | | | | |
| 3.1 | Max Part Weight | 5 | lbf | Desire | Rotary Comparison | Test | Low | This is a static force on the rotary table |
| 3.2A | Average Applied Load | 18 | lbf | Demand | Measured Spindle Loads | Analysis | Medium | Calculated for soft plastics |
| 3.3 | Axis A (Tilt) | 6 | ft-lb | Demand | Hand calculation | Analysis | Medium | Direct Drive torque |
| 3.4 | Axis B (Rotary) | 3 | ft-lb | Demand | Hand calculation | Analysis | Medium | Direct Drive torque |
| 3.5 | Min Natural Frequency | 2000 | Hz | Desire | Hand calculation | Analysis | High | Assume a 4 flute endmill at 30,000 rpm |
| 3.6 | Stiffness | 0.001 | in | Desire | Keep part accuracy | Analysis | High | |
| 4. Energy | | | | | | | | |
| 4.1 | AC Motors OM2 Compatible | - | - | Demand | Compatable with Haas | Test | Low | Motors from Yaskawa |
| 5. Materials | | | | | | | | |
| 5.1 | Rust and corrosion resistant | - | - | Demand | REL Environment | Test | Low | Must function in lubricant |
| 5.2 | Wax machining | - | - | Demand | Sponsor Requirement | Test | Medium | Design to machine wax |
| 6. Signals | | | | | | | | |
| 6.1 | Controlled with 5-axis software | - | - | Demand | G-code | Test | Low | HSM 5th axis software |

| Specification Sheet | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Project 21. 4th and 5th Rotary Stages | | | | | | | | |
| Number | Feature | Value | Unit | Demand vs. Desire | Source | Compliance | Engineering Risk | Notes |
| 6.2 | Simultaneous control with x y z | - | - | Demand | Sponsor Requirement | Test | Medium | |
| 6.2 | 5th axis installation | - | - | Demand | Haas | Test | Low | Use cards provided by Haas |
| 7. Safety | | | | | | | | |
| 7.1 | Handling by trained personel | - | - | Demand | OSHA | Test | Low | Hard point(s) for lifting. Avoid pinch points |
| 7.2 | Safety Factor | 2 | - | Demand | Design safety factor | Analysis | Low | Minimum factor of safety |
| 10. Manufacturing / Production | | | | | | | | |
| 9.1 | Manufactured Part Tolerance | ±0.005 | in | Desire | 0.01in Tolerance | Inspection | High | |
| 9.2 | Part size limitation | 5 | $in^3$ | Demand | Consumer Demand | Inspection | Medium | Cylinder |
| 9.3 | Machining in house | - | - | Desire | Cost | Inspection | Medium | Mustang 60 / IME Dept CNC |
| 11. Assembly | | | | | | | | |
| 10.1 | Instructions | - | - | Demand | Sponsor Requirement | Inspection | Low | Open source documentation |
| 10.2 | Compatible with T-slots | - | - | Demand | Sponsor Requirement | Inspection | Low | Bolts to work table securely |
| 12. Transportation | | | …………………………… | …………………………… | ……………………E-1 | | | |
| 11.1 | Maximum Weight | 40 | lbs | Desire | OM2 Specs + Ergonomics | Inspection | Medium | To be lifted by one person |
| 13. Usage / Operation | | | | | | | | |
| 12.1 | Workholder Bolt Pattern | - | - | Demand | Rotary Comparison | Inspection | Low | To be used with industry work holders |
| 12.2 | Machine lifetime | 2400 | hours | Desire | Estimate | Analysis | High | 3 years, 20 hrs/week, 40 weeks/yr |
| 12.3 | Simultaneous machining | 5 | axes | Demand | Sponsor Requirement | Test | Medium | |
| 13.4 | Homing | - | - | Demand | - | Test | Medium | Must home for each use |
| 13.5 | Operated by trained personel | - | - | Demand | - | Test | Low | User must know how to use CNC |
| 13.6 | Tramming with Square Sides | - | - | Demand | Rotary installation | Inspection | Low | In order to center fixture |
| 14. Maintainance | | | | | | | | |
| 14.1 | Standard Tools/Fluids | - | - | Demand | CNC Maintenance | Inspection | Low | Specialty tools and materials not required |
| 15. Quality Control | | | | | | | | |
| 15.1 | Force Testing | Jan. | month | Desire | Senior Project Timeline | Test | Low | Date: Mid January |
| 15.2 | Functionality Testing | Jan. | month | Desire | Senior Project Timeline | Test | Low | Make a 3" wax impeller |
| 15. Cost | | | | | | | | |
| 15.1 | Max Final Production Cost | 25,000 | USD | Demand | OM2 Cost | Inspection | Medium | Affordable alternative |
| 16. Time Schedule | | | | | | | | |

| Specification Sheet | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Project 21. 4th and 5th Rotary Stages | | | | | | | | |
| **Number** | **Feature** | **Value** | **Unit** | **Demand vs. Desire** | **Source** | **Compliance** | **Engineering Risk** | **Notes** |
| 16.1 | Prototype | Jan. | month | Desire | Senior Project Timeline | Inspection | Low | Delivery Date: January |
| 16.2 | Final Product | May | month | Desire | Senior Project Timeline | Inspection | Low | Delivery Date: May |

# Appendix B

Assembly With Bill of Materials

Housing Detailed Drawing

Shaft Detailed Drawing

Platter Detailed Drawing

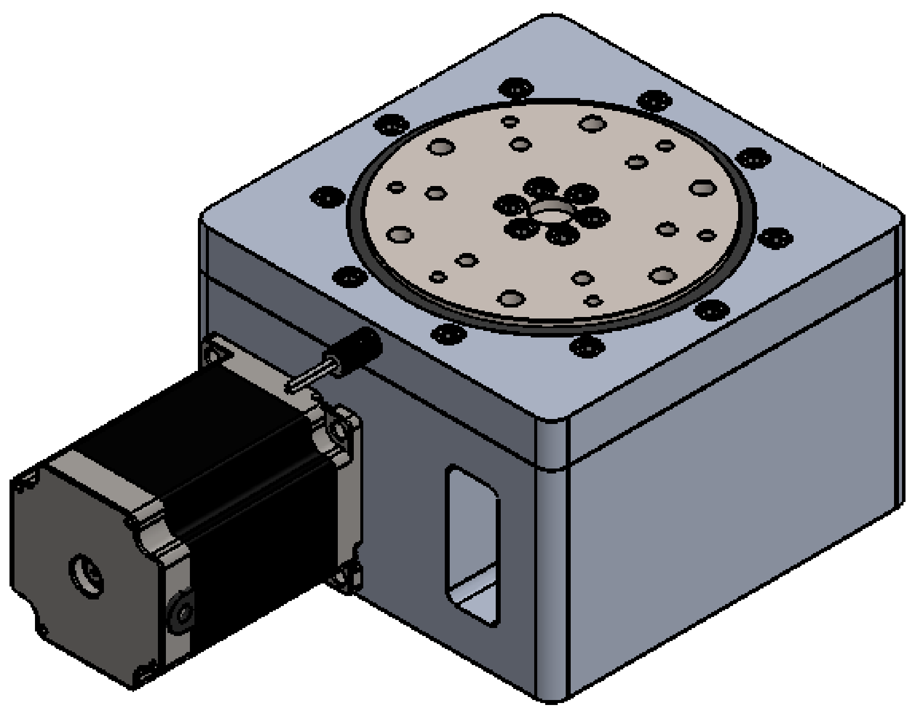
Cover Detailed Drawing

Preload Detailed Drawing

Worm Gear Detailed Drawing

4 3 2 1

B



| ITEM NO. | PART NUMBER | DESCRIPTION | QTY. |
|---|---|---|---|
| 1 | Shaft | | 1 |
| 2 | 6343K16 | Shaft Collar | 1 |
| 3 | 98019A360 | Mil. Spec. Washers | 1 |
| 4 | Platter | | 1 |
| 5 | 97395A475 | Dowel Pins | 2 |
| 6 | 92185A989 | Stainless Steel Socket Head Cap Screws | 11 |
| 7 | HSG | | 1 |
| 8 | 9114m7lmEUL | | 1 |
| 9 | 90201A111 | Extreme-Strength Steel Cap Screws | 1 |
| 10 | 91XPTvqQy3L | | 1 |
| 11 | Cover | | 1 |
| 12 | 9262K409 | Multipurpose O-Rings | 2 |
| 13 | TopGasket | | 1 |
| 14 | 57155K388 | Double Shielded with Extended Inner Ring | 1 |
| 15 | Preload | | 1 |
| 16 | 97395A474 | Dowel Pins | 2 |
| 17 | 3408A21 | Spring plungers | 1 |
| 18 | 2385K46 | | 2 |
| 19 | 65985K22 | Reed Switch | 1 |
| 20 | 65985K22 | Reed Switch | 1 |
| 21 | 6061K411 | Shaft | 1 |
| 22 | BottomGasket | | 1 |
| 23 | Nema 23 Stepper | | 1 |
| 24 | Cap | | 1 |
| 25 | 6138K54 | Ball Bearings | 1 |
| 26 | 6099K21 | Shaft Coupling | 1 |
| 27 | 92185A991 | SS Socket Head Cap Screws | 10 |

A

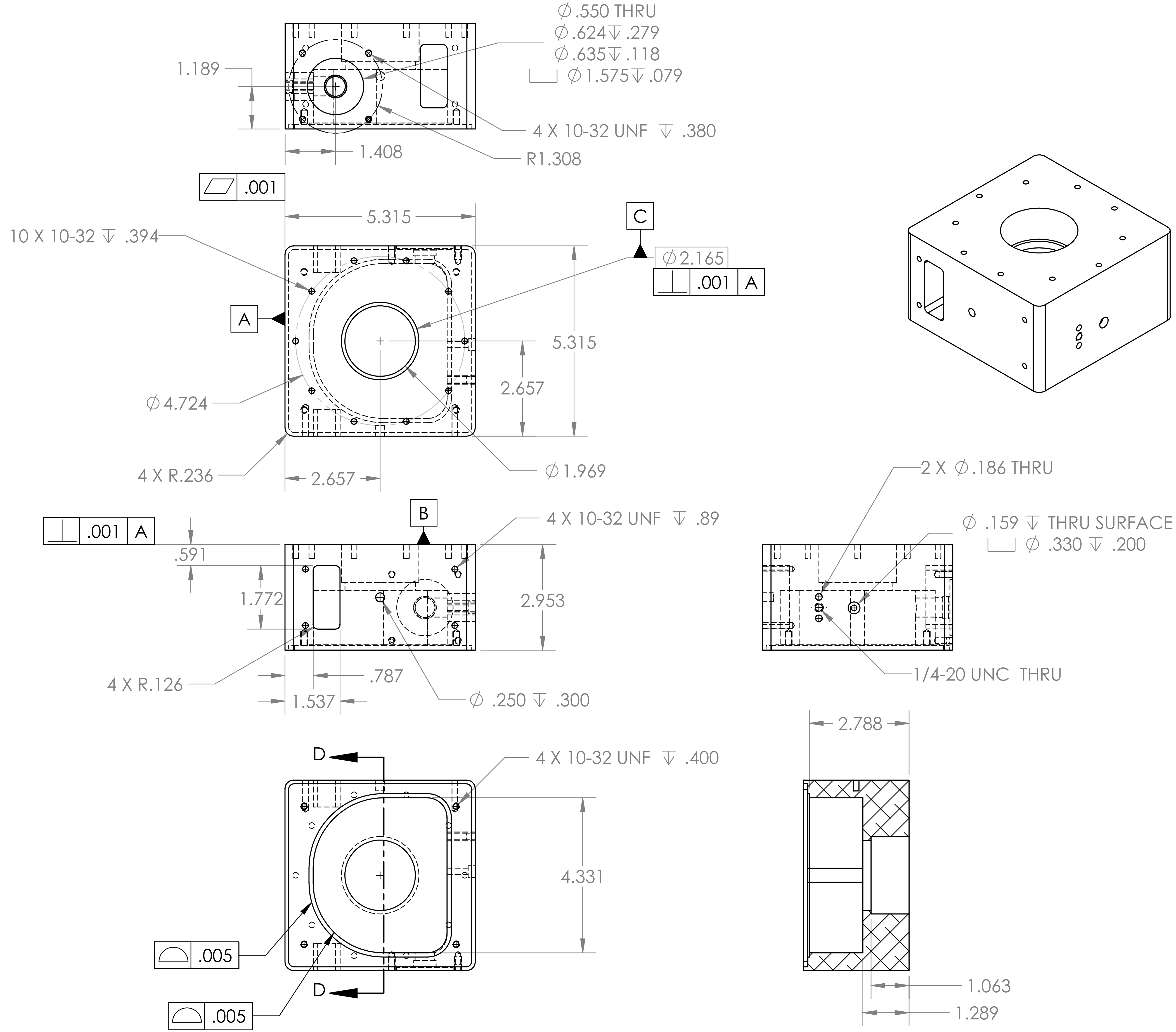
TITLE:

PARTS LIST 5TH AXIS

SIZE **B** DWG. NO. REV

SHEET 1 OF 1

4 3 2 1




SECTION D-D

NOTES

1. NON-DIMENSIONED FEATURES ARE BASIC AND CONTROLLED BY THE MODEL
2. GENERAL PROFILE TOLERANCE APPLIES TO ALL BASIC DIMENSIONS
3. PART TO BE DEBURRED AND CLEANED OF ALL RESIDUE
4. HOLES ARE SPACED EVENLY RADIALLY AROUND THE SPECIFIED RADIUS

Degrees of Freedom

CAL POLY SLO
ME DEPARTMENT

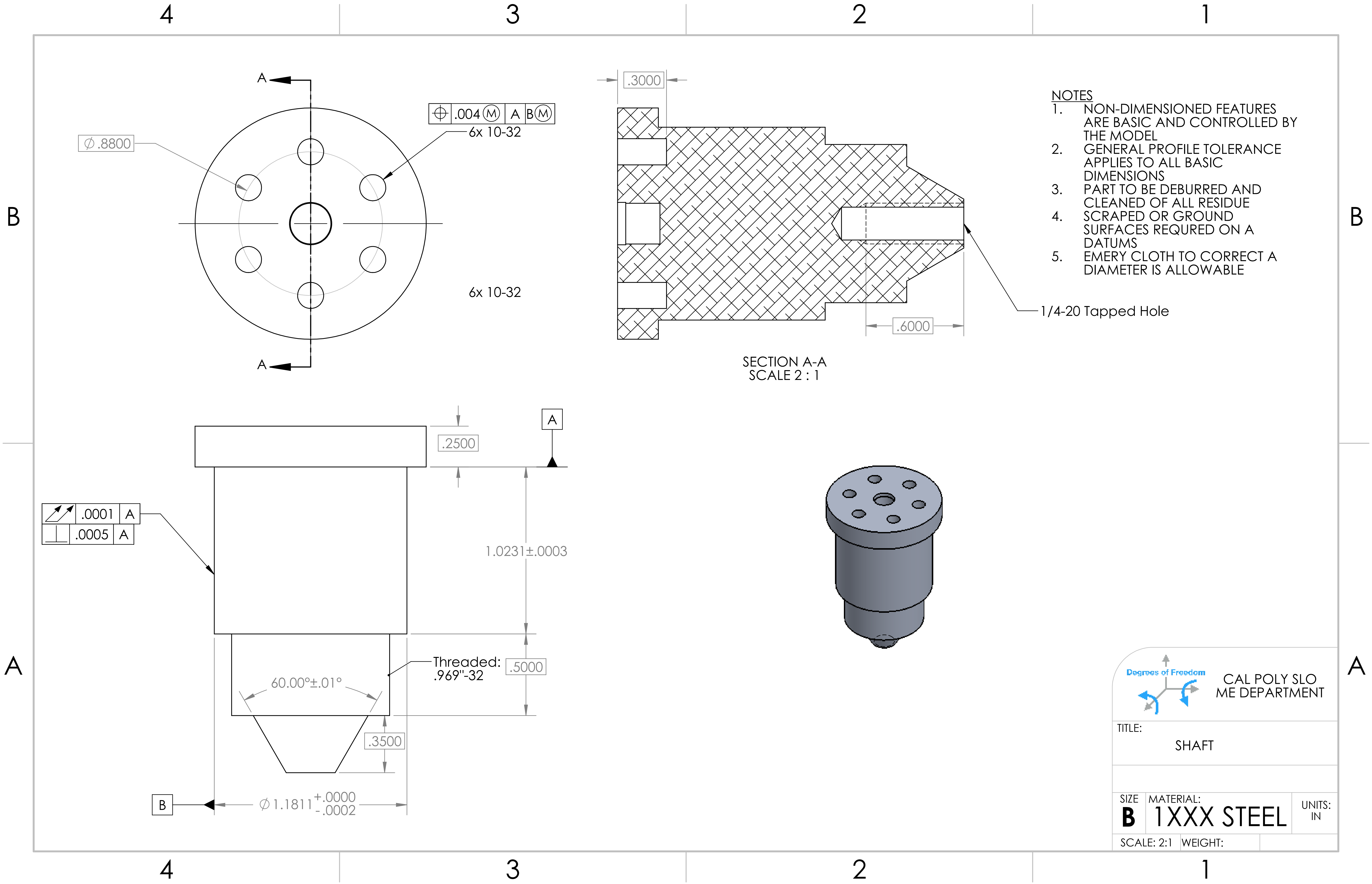
TITLE: HSG B-AXIS

SIZE: **B**

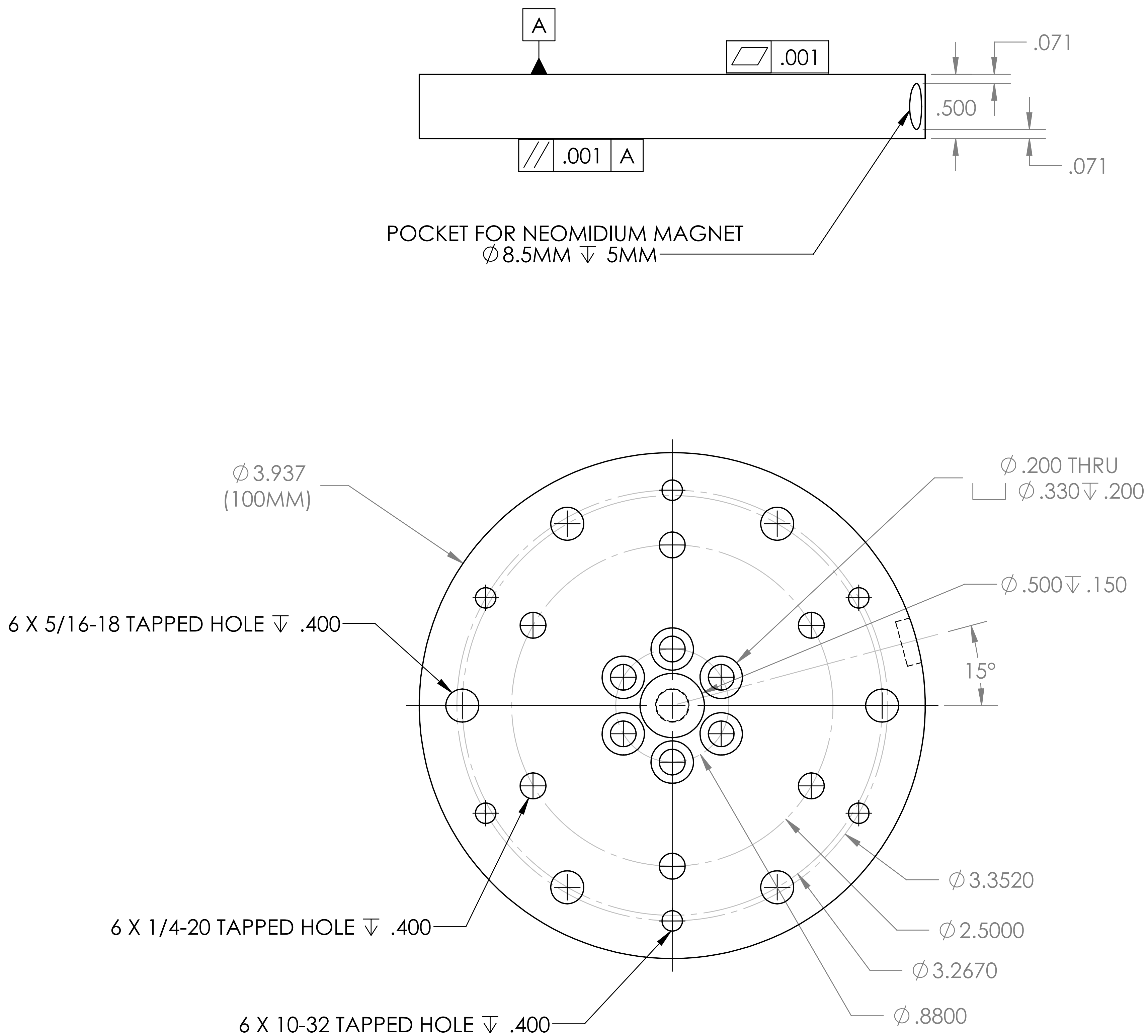
MATERIAL: 6061 AL

UNITS: IN

SCALE: 2:3



4 3 2 1
B
A
A
A
⌀ .8800
.004 Ⓜ A B Ⓜ
6x 10-32
6x 10-32
.3000
.6000
1/4-20 Tapped Hole
SECTION A-A
SCALE 2 : 1
NOTES
1. NON-DIMENSIONED FEATURES ARE BASIC AND CONTROLLED BY THE MODEL
2. GENERAL PROFILE TOLERANCE APPLIES TO ALL BASIC DIMENSIONS
3. PART TO BE DEBURRED AND CLEANED OF ALL RESIDUE
4. SCRAPED OR GROUND SURFACES REQURED ON A DATUMS
5. EMERY CLOTH TO CORRECT A DIAMETER IS ALLOWABLE
.2500
A
.0001 A
.0005 A
1.0231±.0003
Threaded: .969"-32
.5000
60.00°±.01°
.3500
B
⌀ 1.1811 +.0000 -.0002
Degrees of Freedom
CAL POLY SLO
ME DEPARTMENT
TITLE:
SHAFT
SIZE
B
MATERIAL:
1XXX STEEL
UNITS:
IN
SCALE: 2:1
WEIGHT:




NOTES

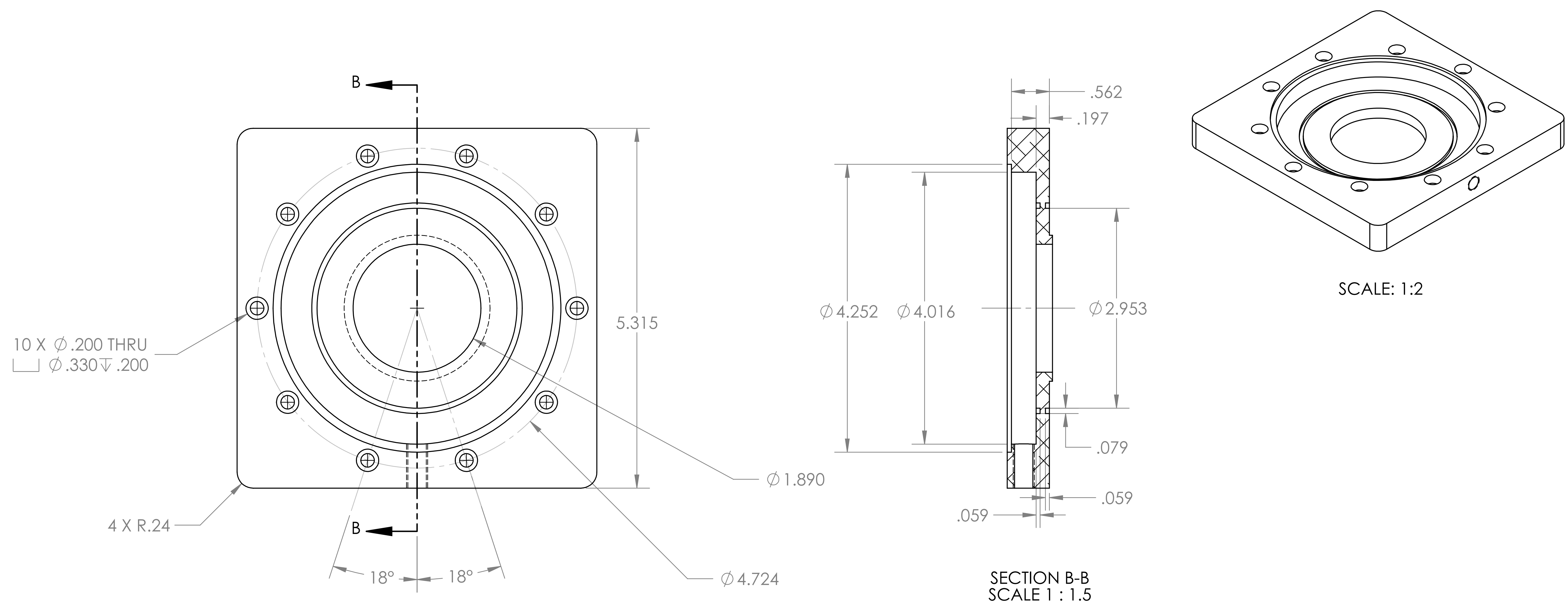
1. NON-DIMENSIONED FEATURES ARE BASIC AND CONTROLLED BY THE MODEL
2. GENERAL PROFILE TOLERANCE APPLIES TO ALL BASIC DIMENSIONS
3. PART TO BE DEBURRED AND CLEANED OF ALL RESIDUE
4. SCRAPED OR GROUND SURFACES REQURED ON ALL DATUMS
5. HOLES ARE SPACED EQUALLY RADIALLY ON EACH SPECIFIED DIAMETER

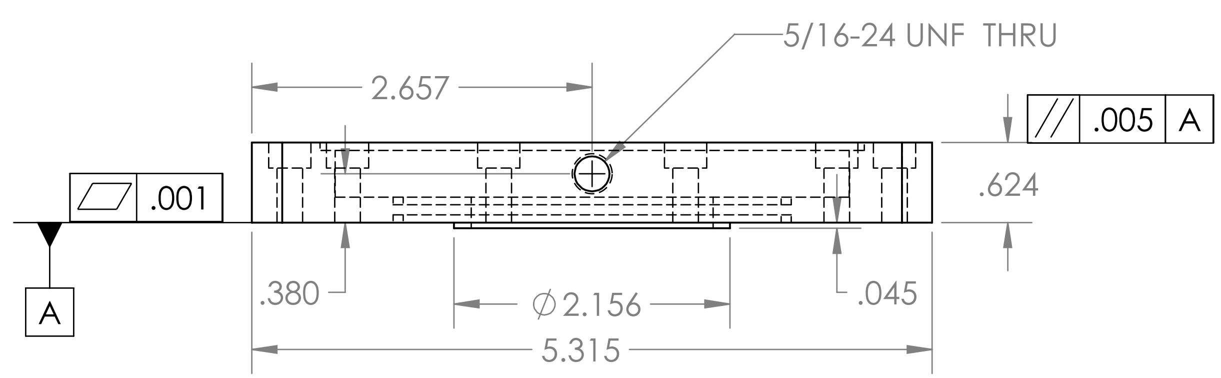
Degrees of Freedom

CAL POLY SLO
ME DEPARTMENT

TITLE:
PLATTER

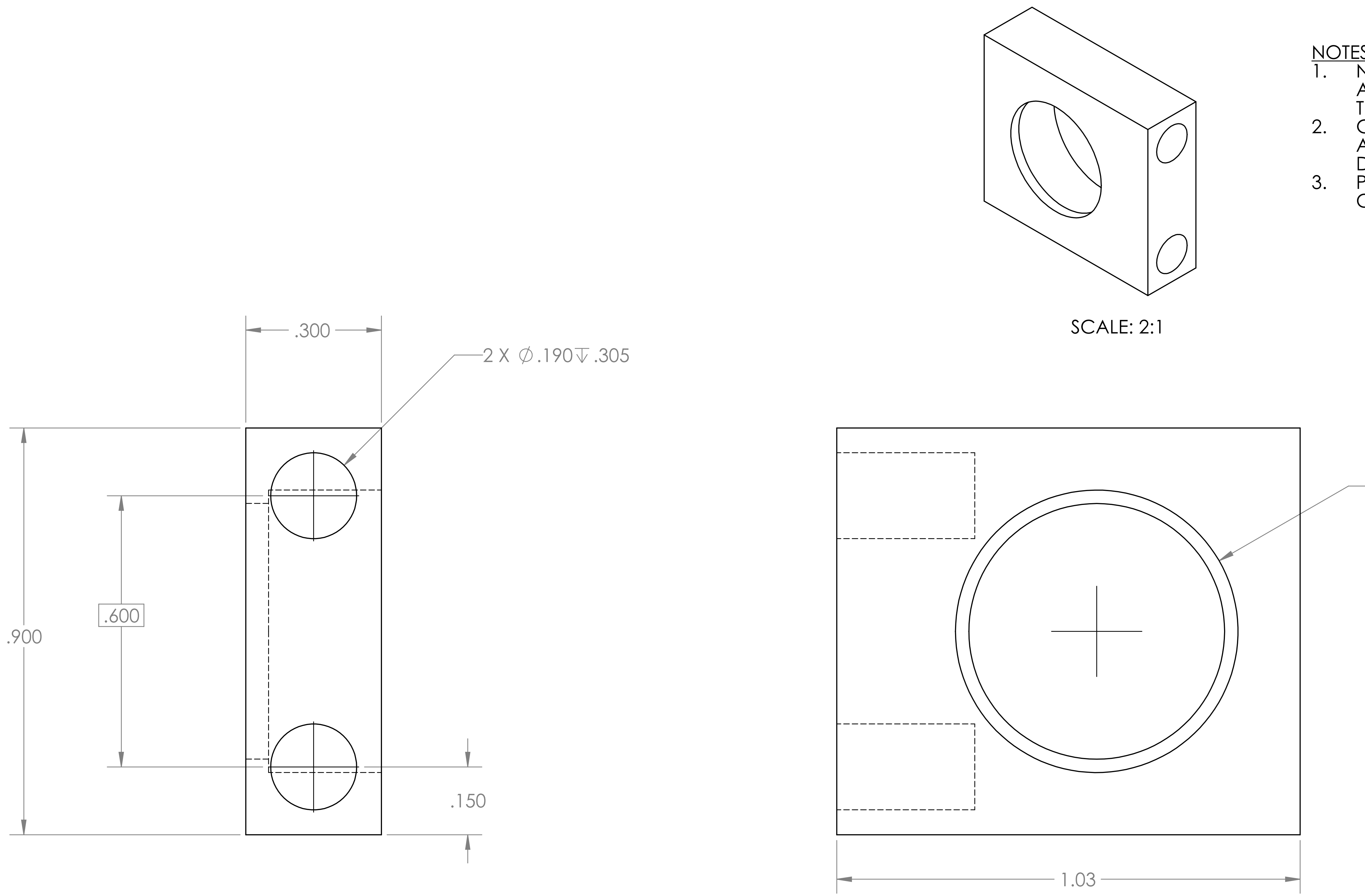
| SIZE | MATERIAL: | UNITS: |
|---|---|---|
| **B** | TOOL STEEL | IN |
| SCALE: 1:1 | WEIGHT: | |




NOTES
1. NON-DIMENSIONED FEATURES ARE BASIC AND CONTROLLED BY THE MODEL
2. GENERAL PROFILE TOLERANCE APPLIES TO ALL BASIC DIMENSIONS
3. PART TO BE DEBURRED AND CLEANED OF ALL RESIDUE
4. HOLES ARE SPACED EVENLY RADIALLY AROUND THE SPECIFIED RADIUS

| | |
|---|---|
| Degrees of Freedom | CAL POLY SLO<br>ME DEPARTMENT |
| TITLE: | CAP |
| SIZE: B | MATERIAL: 6061 AL |
| UNITS: IN | SCALE: 2:3 |



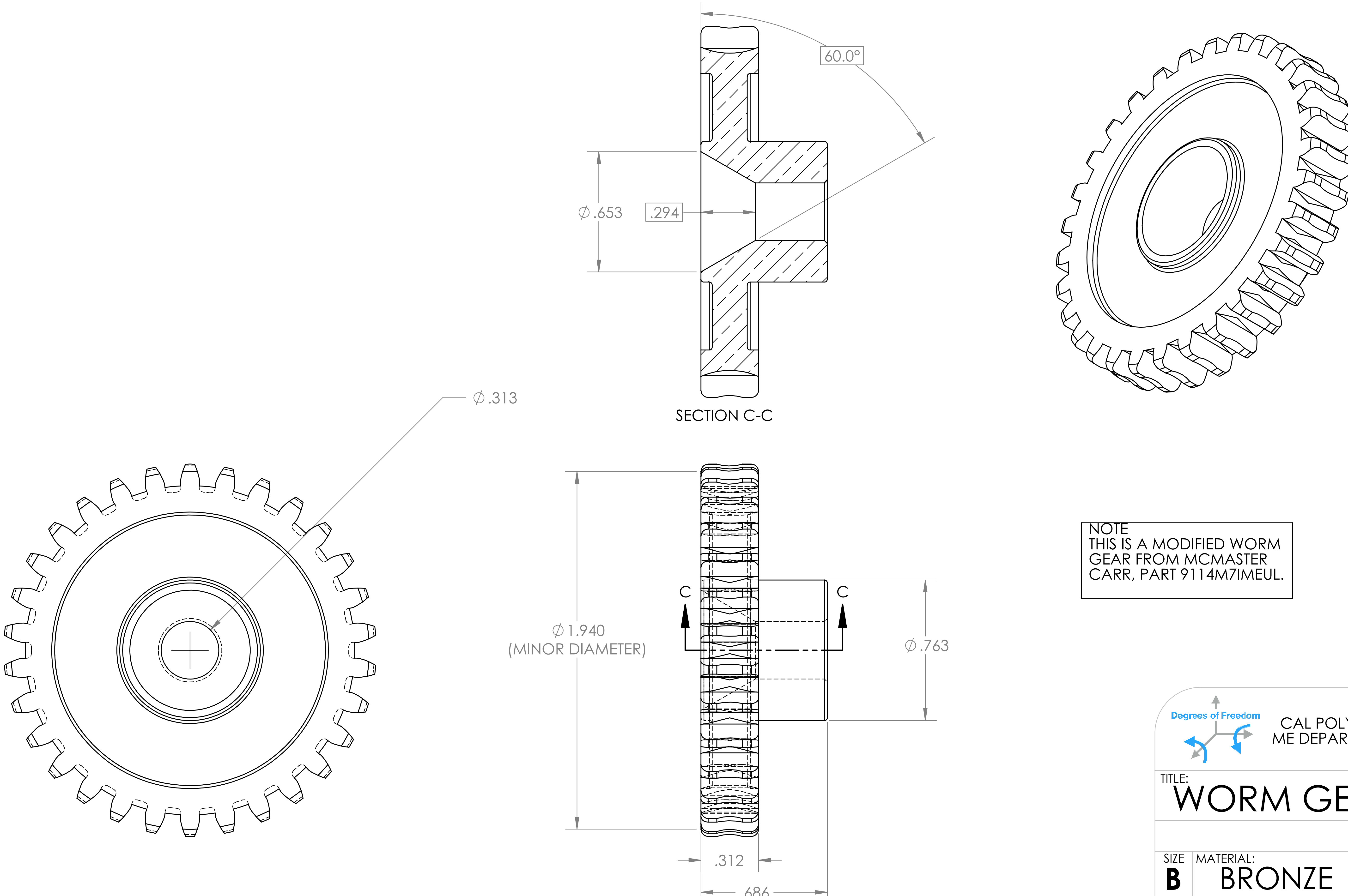

NOTES
1. NON-DIMENSIONED FEATURES ARE BASIC AND CONTROLLED BY THE MODEL
2. GENERAL PROFILE TOLERANCE APPLIES TO ALL BASIC DIMENSIONS
3. PART TO BE DEBURRED AND CLEANED OF ALL RESIDUE

Degrees of Freedom
CAL POLY SLO
ME DEPARTMENT

TITLE: SPRING FORCER

| SIZE | MATERIAL: | UNITS: |
|---|---|---|
| B | 6061 AL | IN |

SCALE: 2:3




NOTE
THIS IS A MODIFIED WORM GEAR FROM MCMASTER CARR, PART 9114M7IMEUL.

Degrees of Freedom

CAL POLY SLO
ME DEPARTMENT

TITLE:
# WORM GEAR

SIZE: B

MATERIAL: BRONZE

UNITS: IN

SCALE: 2:1

# Appendix C

Structured Bill of Materials

Cost Bill of Materials

## Structured Bill of Materials

| PART # | | | | | QTY | PART | FUNCTION | DRAWING # | SUPPLIER | SUPPLIER # |
|---|---|---|---|---|---|---|---|---|---|---|
| L1 | L2 | L3 | L4 | L5 | | | | | | |
| 1 | | | | | 1 | Rotary Table | Overall | 10000 | | |
| | 1 | | | | 1 | Axis B | One axis of Rotation | 11000 | | |
| | | 1 | | | 1 | Housing Body Sub-Assembly | Base of one rotary axis | - | - | - |
| | | | 1 | | 1 | Housing Body | Holds shaft and gear train assem. | 11110 | MAKE | - |
| | | | 2 | | 1 | 14mm Ball bearing | Locates worm shaft | - | McMaster | 2423K24 |
| | | | 3 | | 2 | 0.25"x0.5" Dowel Pins | Locates attachment plate onto housing | - | McMaster | 97395A475 |
| | | 2 | | | 1 | Platter Sub-Assembly | Holds working part | - | - | - |
| | | | 1 | | 1 | Platter | Holds working part | 11210 | MAKE | - |
| | | | 2 | | 6 | 10-32 x 0.5" Socket Head Cap Screws | Tightens platter onto cap | - | McMaster | 92185A989 |
| | | 3 | | | 1 | Cap Sub-Assembly | Retains bearing race | - | - | - |
| | | | 1 | | 1 | Cap | Retains bearing race | 11310 | MAKE | - |
| | | | 2 | | 1 | Gasket | Prevents material from entering housing | 11320 | McMaster | 8837K112 |
| | | | 3 | | 10 | 10-32 x 0.75" Socket Head Cap Screws | Tightens cap onto housing | - | McMaster | 96006A693 |
| | | | 4 | | 2 | 75mm x 3mm wide O-ring | Prevents oil from entering housing | - | McMaster | 9262K409 |
| | | | 5 | | 1 | 0.25" x 0.5" Dowel Pin | Locates center of cap to platter | - | McMaster | 97395A475 |
| | | | 6 | | 1 | SPDT 150V DC Reed Switch | For homing | - | McMaster | 6585K22 |
| | | 4 | | | | Shaft Sub-Assembly | Allows work piece to rotate | - | - | - |
| | | | 1 | | 1 | Shaft | Holds bearings and platter | 11410 | MAKE | - |
| | | | 2 | | 2 | 7006C NSK Bearings | Allows platter to rotate in housing | - | Mcmaster | 2385K46 |
| | | | 3 | | 1 | 1-9/16"Bearing Lock Nut | Preloads bearings | - | McMaster | 6343K160 |
| | | 5 | | | 1 | Gear Train Sub-Assembly | Turns motor rotation into mech power | - | | |
| | | | 1 | | 1 | 30T 16DP Gear | Rotates shaft, gives 30:1 reduction | - | Boston Gear | G1043 |
| | | | 2 | | 1 | 16DP Worm | Contacts 30T gear | - | Boston Gear | LVHB1 |
| | | | 3 | | 1 | 3mm Set Screw | Holds worm in place | - | McMaster | 92015A101 |
| | | | 4 | | 1 | 1/4" Shaft | Rotates worm, attaches to motor | - | McMaster | 1144K11 |
| | | | 5 | | 1 | 1/4" ID Washer | Evens out force over gear, prevents loosening | - | McMaster | 98019A360 |
| | | | 6 | | 1 | 3/4"x.75 Cap Screw | Attaches 30T gear to shaft | - | McMaster | 90201A111 |
| | | | 7 | | 1 | Gear Preload Sub-Assembly | Preloads gears to prevent major backlash | - | - | - |
| | | | | 1 | 1 | Gear Preload Mechanism | Pushes worm against gear | 11571 | MAKE | - |
| | | | | 2 | 1 | 1/4" Ball Bearing | Locates worm shaft | - | McMaster | 57155K388 |
| | | | | 3 | 1 | Ball-Nose Spring Plunger | Constant force engagement on worm | - | McMaster | 3408A75 |
| | | | | 4 | 2 | 3/16" x 1.25" Dowel Pins | Locates Gear preload mechanism | | McMaster | 97395474 |
| | | 6 | | | 1 | Motor Assembly | Provides power to table | - | | |
| | | | 1 | | 1 | 100W Motor | Rotates worm shaft | - | Yaskawa | SGMJV-01A |
| | | | 2 | | 1 | 1/4"-8mm Shaft Coupling | Connects motor shaft to worm shaft | - | McMaster | 2764K123 |
| | | | 3 | | 1 | Motor Cable | Powers motor | - | Yaskawa | - |
| | | | | | | | | | | |
| | 2 | | | | 1 | Axis A | Rotates 4th Axis | 12000 | | |
| | | 1 | | | 1 | Housing Body Sub-Assembly | Base of one rotary axis | - | - | - |
| | | | 1 | | 1 | Housing Body | Holds shaft and gear train assem. | 11110 | MAKE | - |
| | | | 2 | | 1 | 14mm Ball bearing | Locates worm shaft | - | McMaster | 2423K24 |
| | | 2 | | | 1 | Platter Sub-Assembly | Holds working part | - | - | - |
| | | | 1 | | 1 | Platter | Holds working part | 11210 | MAKE | - |
| | | | 2 | | 6 | 10-32 x 0.5" Socket Head Cap Screws | Tightens platter onto cap | - | McMaster | 92185A989 |
| | | | 3 | | 2 | 0.25"x0.5" Dowel Pins | Locates attachment plate onto Axis | - | McMaster | 97395A475 |
| | | 3 | | | 1 | Cap Sub-Assembly | Retains bearing race | - | - | - |
| | | | 1 | | 1 | Cap | Retains bearing race | 11310 | MAKE | - |
| | | | 2 | | 1 | Gasket | Prevents material from entering housing | - | McMaster | 8837K112 |
| | | | 3 | | 10 | 10-32 x 0.75" Socket Head Cap Screws | Tightens cap onto housing | - | McMaster | 96006A693 |
| | | | 4 | | 2 | 75mm x 3mm wide O-ring | Prevents oil from entering housing | - | McMaster | 9262K409 |
| | | | 5 | | 1 | 0.25" x 0.5" Dowel Pin | Locates center of cap to platter | - | McMaster | 97395A475 |
| | | 4 | | | | Shaft Sub-Assembly | Allows work piece to rotate | - | - | - |
| | | | 1 | | 1 | Shaft | Holds bearings and platter | - | MAKE | - |
| | | | 2 | | 2 | 7006C NSK Bearings | Allows platter to rotate in housing | - | McMaster | 2385K46 |
| | | | 3 | | 1 | 1-9/16"Bearing Lock Nut | Preloads bearings | - | McMaster | 6343K160 |
| | | 5 | | | 1 | Gear Train Sub-Assembly | Turns motor rotation into mech power | - | | |
| | | | 1 | | 1 | 30T 16DP Gear | Rotates shaft, gives 30:1 reduction | - | Boston Gear | G1043 |
| | | | 2 | | 1 | 16DP Worm | Contacts 30T gear | - | Boston Gear | LVHB1 |
| | | | 3 | | 1 | 3mm Set Screw | Holds worm in place | - | McMaster | 92015A101 |
| | | | 4 | | 1 | 1/4" Shaft | Rotates worm, attaches to motor | 11540 | McMaster | 1144K11 |
| | | | 5 | | 1 | 1/4" ID Washer | Evens out force over gear, prevents loosening | - | McMaster | 98019A360 |
| | | | 6 | | 1 | 3/4"x.75 Cap Screw | Attaches 30T gear to shaft | - | McMaster | 90201A111 |
| | | | 7 | | 1 | Gear Preload Sub-Assembly | Preloads gears to prevent major backlash | - | - | - |
| | | | | 1 | 1 | Gear Preload Mechanism | Pushes worm against gear | 11571 | MAKE | - |
| | | | | 2 | 1 | 1/4" Ball Bearing | Locates worm shaft | - | McMaster | 57155K388 |
| | | | | 3 | 1 | Ball-Nose Spring Plunger | Constant force engagement on worm | - | McMaster | 3408A75 |
| | | | | 4 | 2 | 3/16" x 1.25" Dowel Pins | Locates Gear preload mechanism | | McMaster | 97395474 |
| | | 6 | | | 1 | Motor Assembly | Provides power to table | - | | |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 1 | | 1 | 200W Motor | Rotates worm shaft | - | Yaskawa | SGMJV-02A |
| | | | 2 | | 1 | 1/4"-14mm Shaft Coupling | Connects motor shaft to worm shaft | - | McMaster | 2764K322 |
| | | | 3 | | 1 | Motor Cable | Powers motor | - | Yaskawa | - |
| | | | | | | | | | | |
| | 3 | | | | 1 | Attachment Plate Sub-Assembly | Combines axes | - | - | - |
| | | 1 | | | 1 | Fixture Plate | Allows Axis B to attach to A | 13100 | MAKE | - |
| | | 2 | | | 8 | 1/4"-20 Socket Head bolts | Attaches fixture plate to both axes | - | McMaster | 90128A242 |
| | | | | | | | | | | |
| | 4 | | | | 1 | Gear Pocket End Cap Sub-Assembly | Seals opening in housing | - | - | - |
| | | 1 | | | 1 | Gear Pocket Gasket | Prevents oil from leaking out of housing | 14100 | McMaster | 8837K112 |
| | | 2 | | | 1 | Gear Pocket Lid | Closes opening in housing | 14200 | - | - |
| | | 3 | | | 4 | 10-32 x 0.75" Socket Head Cap Screws | Tightens cap onto housing | - | McMaster | 96006A693 |

| Cost Analysis | | | | |
|---|---|---|---|---|
| **Part/Material** | **Qty** | **Supplier** | **Supplier #** | **Cost** |
| Steel | 1 | IME Dept | - | N/A |
| 7006C NSK Bearings | 4 | McMaster | 2385K46 | 1,091.00 |
| 8mm Ball bearing | 1 | McMaster | 2423K21 | 15.15 |
| 14mm Ball bearing | 1 | McMaster | 2423K24 | 19.11 |
| 1/4"-8mm Shaft Coupling | 1 | McMaster | 2764K123 | 60.54 |
| 1/4"-14mm Shaft Coupling | 1 | McMaster | 2764K322 | 78.69 |
| Ball-Nose Spring Plunger | 2 | McMaster | 3408A75 | 7.24 |
| 1/4" Ball Bearing | 2 | McMaster | 57155K388 | 9.56 |
| 1/4" Shaft | 4 | McMaster | 6061K411 | 4.20 |
| 1-9/16"Bearing Lock Nut | 2 | McMaster | 6343K160 | 14.44 |
| SPDT 150V DC Reed Switch | 2 | McMaster | 6585K22 | 92.22 |
| Gasket Material - Neoprene | 2 | McMaster | 8837K112 | 21.70 |
| 1/4"-20 Socket Head bolts | 8 | McMaster | 90128A242 | 9.09 |
| 1/4"-20x.75 Cap Screw | 2 | McMaster | 90201A111 | 9.62 |
| 3mm Set Screw | 2 | McMaster | 92015A101 | 9.34 |
| 10-32 x 0.5" Socket Head Cap Screws | 13 | McMaster | 92185A989 | 4.26 |
| 50mm x 2mm O-ring | 2 | McMaster | 9262K208 | 9.31 |
| 75mm x 2mm wide O-ring | 4 | McMaster | 9262K409 | 6.26 |
| 10-32 x 0.75" Socket Head Cap Screws | 24 | McMaster | 96006A693 | 5.71 |
| 3/16" X 1 1/8" Dowel Pins | 2 | McMaster | 97395A474 | 11.25 |
| 0.25"x0.5" Dowel Pins | 6 | McMaster | 97395A475 | 10.21 |
| 1/4" ID Washer | 2 | McMaster | 98019A360 | 6.75 |
| 30T 16DP Gear | 2 | Boston Gear | G1043 | 91.60 |
| 16DP Worm | 2 | Boston Gear | LVHB1 | 46.40 |
| 100W Motor + Cable | 1 | Yaskawa | SGMJV-01A | 500.00 |
| 200W Motor + Cable | 1 | Yaskawa | SGMJV-02A | 550.00 |
| Aluminum | 1 | McMaster | | 290.00 |
| | | | | |
| **Other Items** | | | | |
| Emery Cloth | 1 | McMaster | 8238A53 | 9.93 |
| Cylinder Hone | 1 | Ebay | - | 20.00 |
| Spanner Wrench | 1 | McMaster | 6975A16 | 19.58 |
| 0.001" Shim Stock | 4 | McMaster | 97022A151 | 30.32 |
| | | | **Total** | 3,053.48 |
| | | | **Actual** | 2,003.48 |

# Appendix D

¾ in Cap Screw

3mm Set Screw

10-32x0.5” Cap Screw

10-32x0.75” Cap Screw

Bearing Lock Nut

Dowel Pin

Gasket Rubber

Linear Ball Bearing

Platter O-Ring

Washer

16DP Worm

30 Tooth, 16DP Gear

Worm and Gear Combined

Worm Shaft

¼ in Ball Bearing

NSK Angular Contact Ball Bearings

¼ in to 14mm Shaft Coupling

¼ in to 8mm Shaft Coupling

Ball-Nose Spring Plunger

¼”-20x0.25” Socket Head Cap Screw

3/16”x1.75” Dowel Pin

SPDT 150V DC Reed Switch

14mm Ball Bearing

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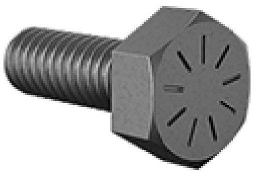
Text 75930

# Extreme-Strength Grade 9 Steel Cap Screw

## 1/4"-20 Fully Threaded, 3/4" Long, Zinc-Plated

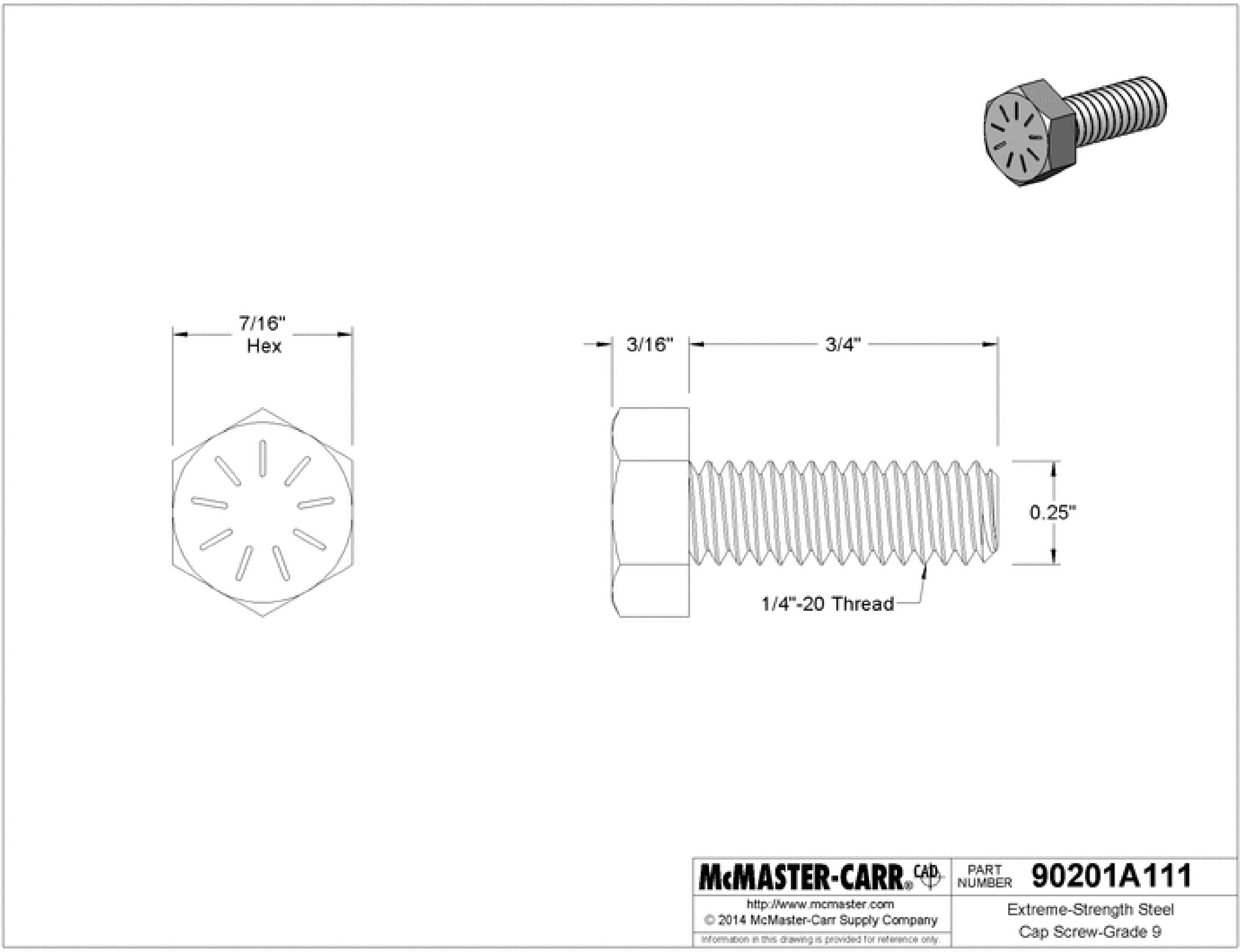
In stock
$9.62 per pack of 25
90201A111



| | |
|---|---|
| Material | Steel |
| Grade | 9 |
| Finish | Zinc Yellow-Chromate Plated |
| Thread Size | 1/4"-20 |
| Head Width | 7/16" |
| Head Height | 3/16" |
| Screw Size | 1/4" (0.250") |
| Length | 3/4" |
| Thread Length | Full |
| RoHS | Not Compliant |

These are our strongest cap screws. They are made from alloy steel. Length is measured from under the head.

Inch screws have a hefty head that's 20% taller than a standard cap screw head and a minimum tensile strength of 180,000 psi. They are zinc yellow-chromate plated for rust resistance and have nine radial markings to indicate Grade 9. They have a minimum Rockwell hardness of C38 and a Class 2A thread fit. Use with Grade 9 nuts and washers.




The information in this 3-D model is provided for reference only.

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# 18-8 Stainless Steel Cup Point Set Screw

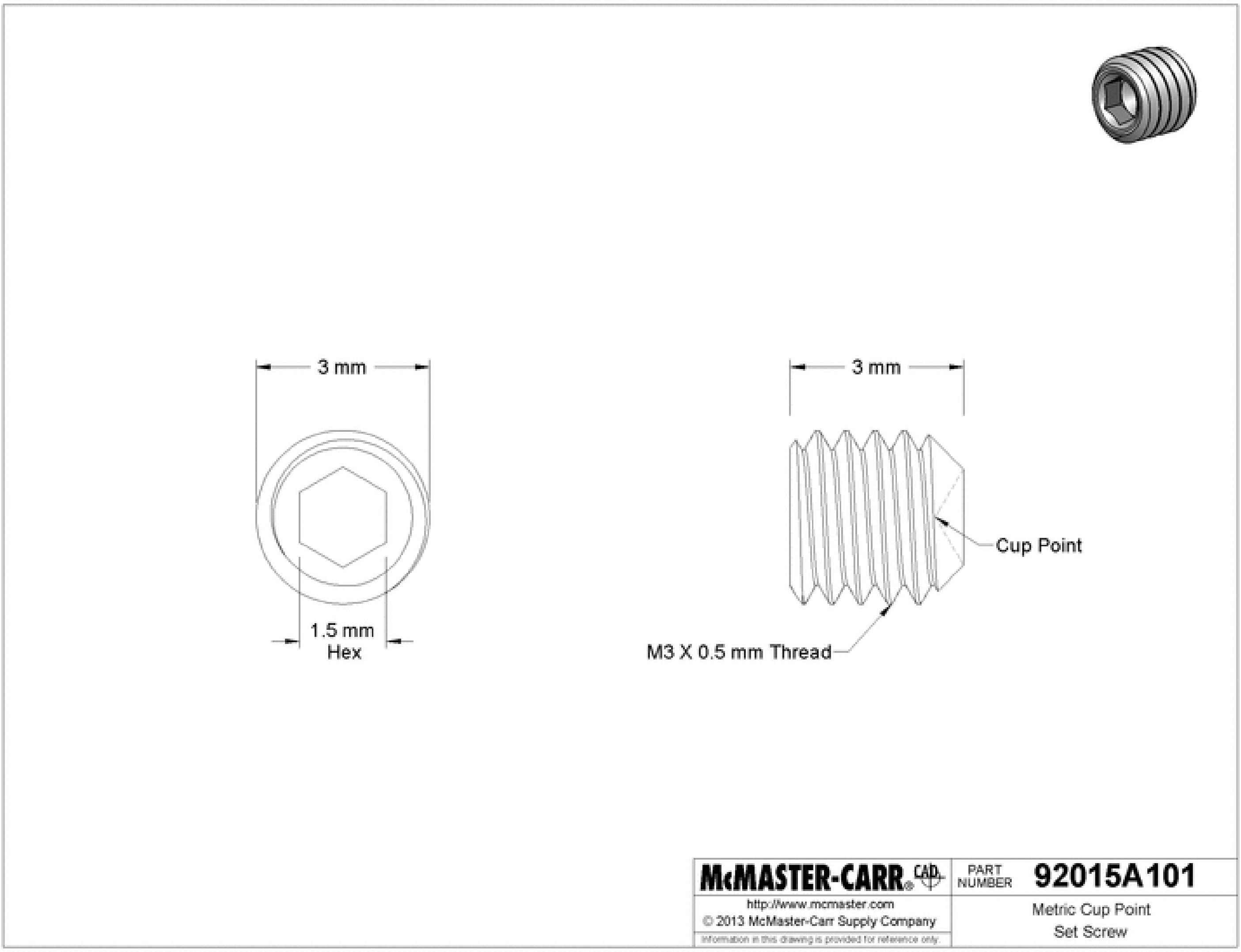
## M3 Size, 3mm Long, 0.5mm Pitch

In stock
$9.34 per pack of 100
92015A101

| | |
|---|---|
| Point Style | Cup |
| Material | 18-8 Stainless Steel |
| Thread Size | M3 |
| Pitch | 0.5 mm |
| Length | 3 mm |
| Drive Style | Hex Socket |
| Hex Size | 1.5 mm |
| RoHS | Compliant |

Screws have a hex socket. Length listed is the overall length.

Metric screws meet DIN 916/ISO 4029. Stainless steel screws have a Class 6g thread fit.

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# Type 316 Stainless Steel Socket Head Cap Screw

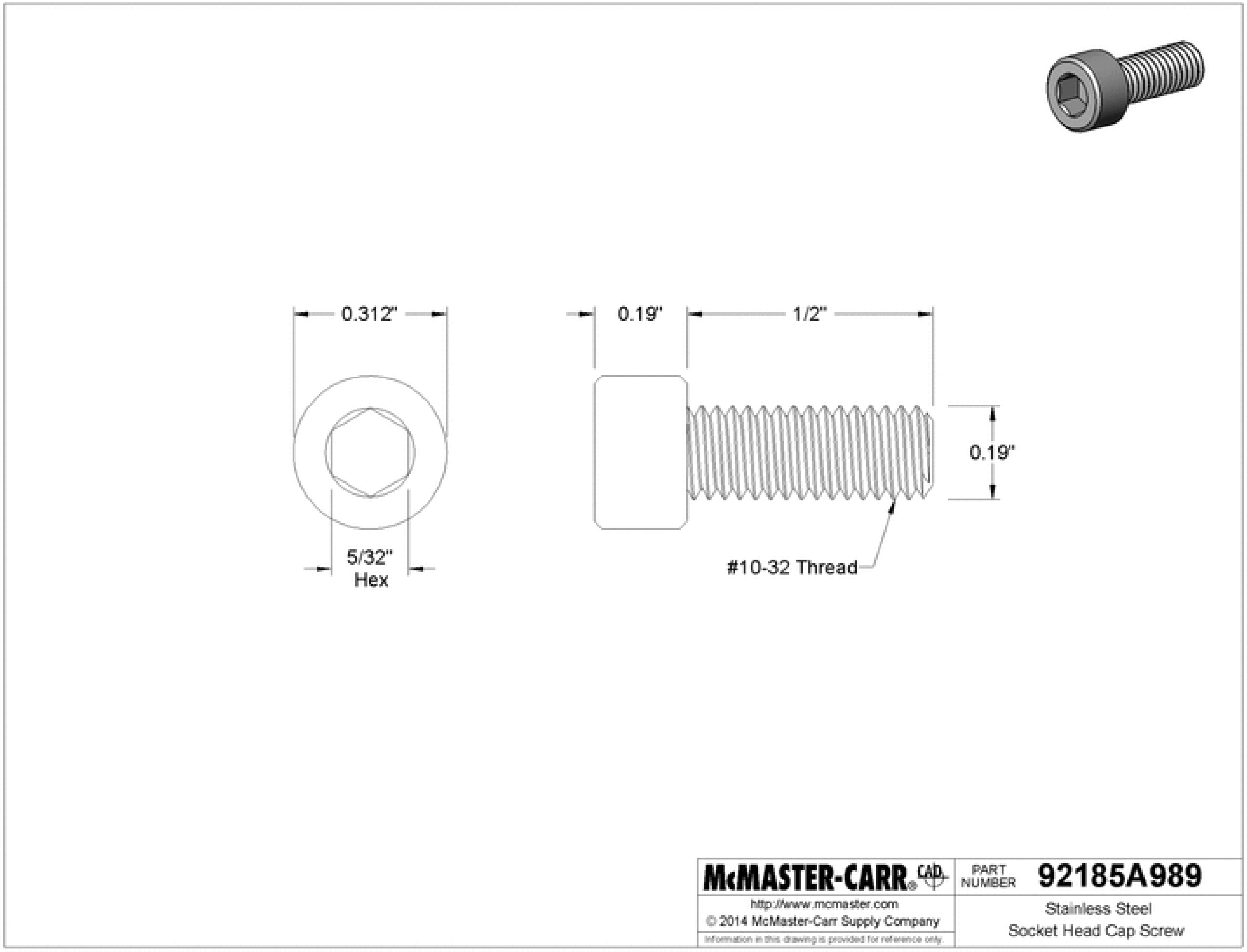
## 10-32 Thread, 1/2" Length

In stock
$4.26 per pack of 25
92185A989

| | |
|---|---|
| Thread Size | 10-32 |
| Length | 1/2" |
| Thread Length | Full |
| Additional Specifications | Type 316 Stainless Steel |
| RoHS | Compliant |

Choose stainless steel screws when you need corrosion resistance. Length is measured from under the head.

Inch screws have a Class 3A thread fit, a minimum tensile strength of 70,000 psi, and a minimum Rockwell hardness of B70.

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# Black Oxide 18-8 Stainless Steel Socket Head Cap Screw

## 10-32 Thread, 3/4" Length

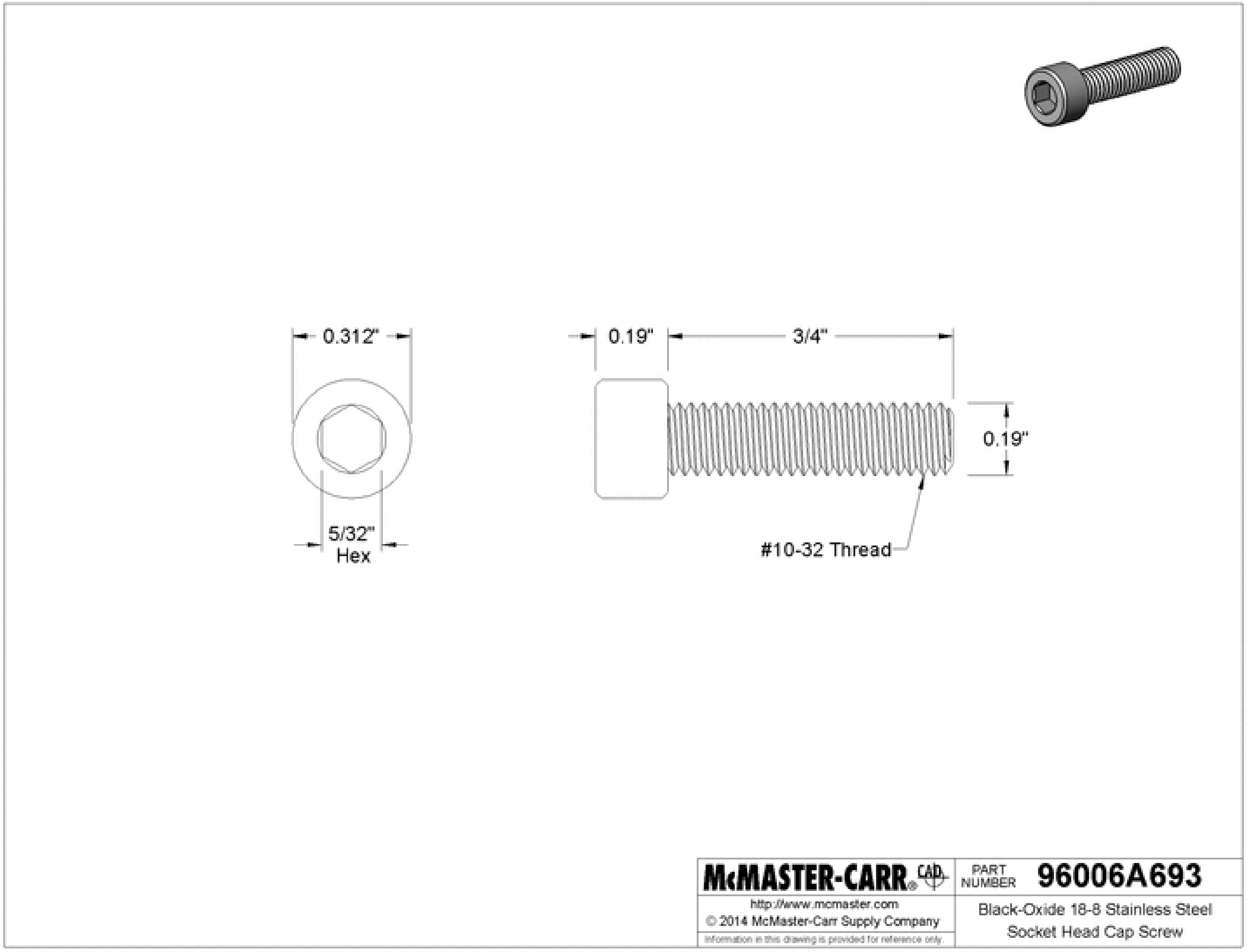
In stock
$5.71 per pack of 25
96006A693

| | |
|---|---|
| Thread Size | 10-32 |
| Length | 3/4" |
| Thread Length | Full |
| Additional Specifications | Black-Oxide 18-8 Stainless Steel |
| RoHS | Compliant |

Choose stainless steel screws when you need corrosion resistance. Length is measured from under the head.

Inch screws have a Class 3A thread fit, a minimum tensile strength of 70,000 psi, and a minimum Rockwell hardness of B70.

Black Oxide—Screws have a matte-black finish.

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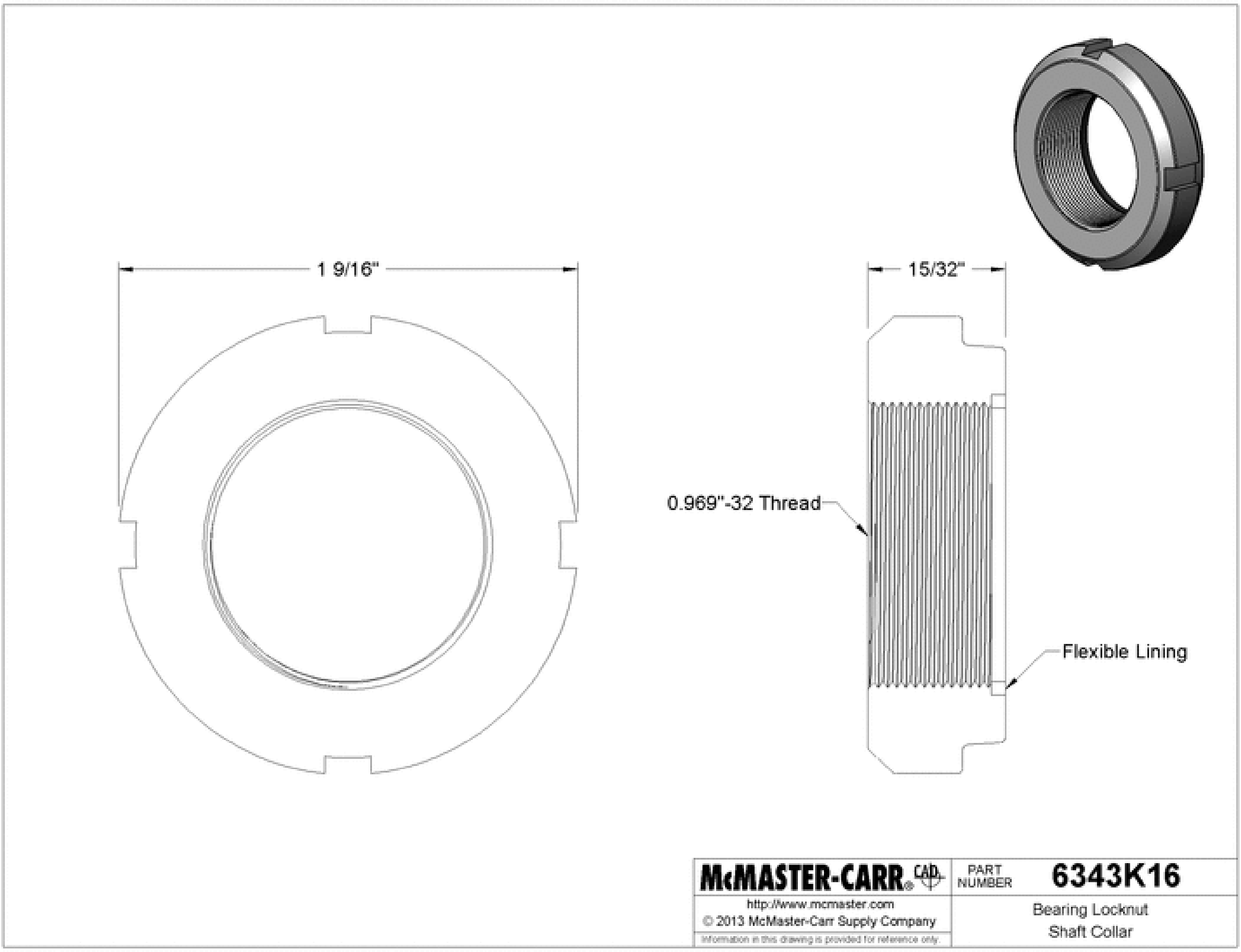
# Bearing Locknut Shaft Collar
## with 0.969"-32 Thread, Steel

In stock
$7.22 Each
6343K16

| | |
|---|---|
| Material | Steel |
| Inch/Metric | Inch |
| Thread Size | 0.969"-32 |
| OD | 1 9/16" |
| Width | 15/32" |

In addition to fine threads, which provide close engagement on your shaft, collars have a flexible lining that hugs your shaft to lock your bearing in place. They help eliminate the looseness commonly associated with locknuts that use keyways and lock washers, which are not required with these collars. They also allow precise positioning and better squareness of the retaining ring to the bearing face to hold bearings in place. All collars work with SAE, ABEC, and ANSI/ABMA standard products.

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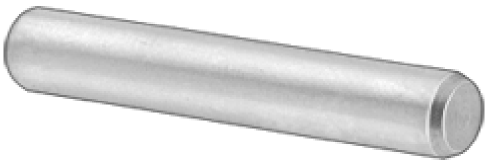
# Corrosion Resistant Dowel Pin

## Type 316 Stainless Steel, 1/4" Diameter, 1/2" Length

In stock
$10.21 per pack of 10
97395A475

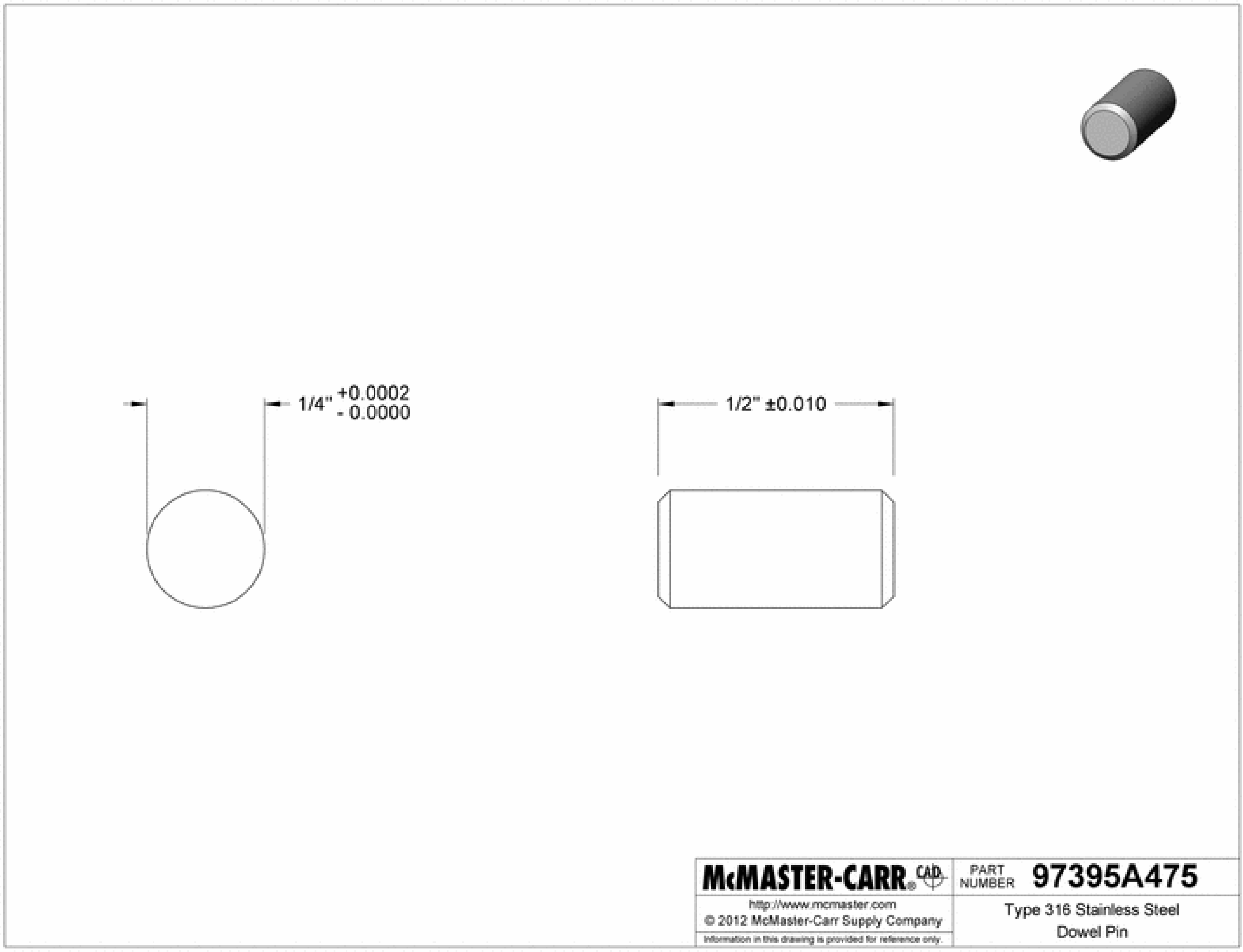


| | |
|---|---|
| Length | 1/2" |
| Additional Specifications | Type 316 Stainless Steel<br>1/4" Dia. |
| RoHS | Compliant |

Pins are commonly used as pivots, hinges, shafts, jigs, and fixtures to locate or hold parts. They are precision ground (unless noted) for more accurate alignment, and slightly oversized for a tight fit. To aid insertion, ends are beveled or rounded (unless noted). For a tight fit, your hole should be equal to or slightly smaller than the diameter shown. Breaking strength and Rockwell hardness are not rated (unless noted). Materials that have a breaking strength are measured as double shear, which is the force required to break a pin into three pieces.

Type 316 Stainless Steel—Our most corrosion resistant dowel pins. May be magnetic. Diameter tolerance is +0.0002".

Not rated for breaking strength.

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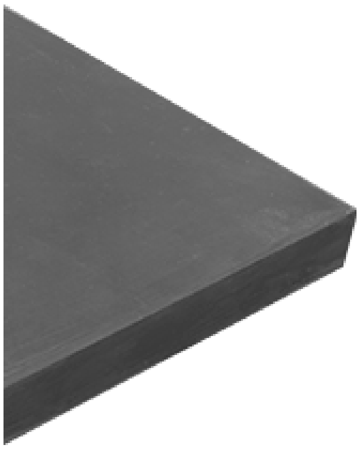
# Nonmarking Neoprene Rubber

## 1/16" Thick, 12" x 12", 50 Durometer, Black

In stock
$10.85 Each
8837K112



| | |
|---|---|
| Shape | Sheet |
| Material | Neoprene |
| Texture | Smooth |
| Thickness | 1/16" |
| Thickness Tolerance | -0.016" to +0.016" |
| Width | 12" |
| Width Tolerance | +0.500" |
| Length | 12" |
| Length Tolerance | +0.5" |
| Backing Type | Plain |
| For Use Outdoors | No |
| Temperature Range | -30° to 220° F |
| Tensile Strength | 1,200 psi |
| Color | Black |
| Specifications Met | ASTM D2000 BC |
| Durometer | 50A (Medium) |
| Durometer Tolerance | -5 to +5 |
| RoHS | Compliant |

While other black rubber can leave a mark after use, these specially formulated neoprene sheets leave no evidence behind. Neoprene is also known as chloroprene. It has good oil and abrasion resistance.

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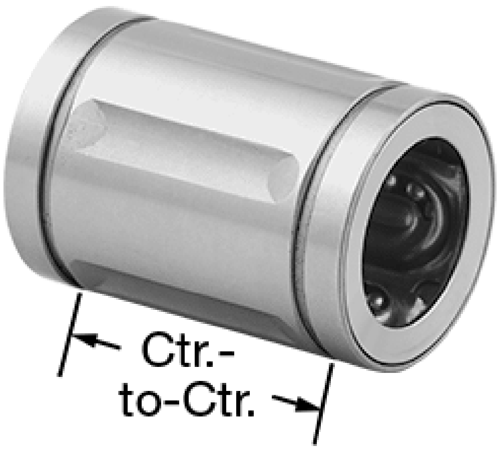
# Fixed Alignment High-Temperature Linear Ball Bearing

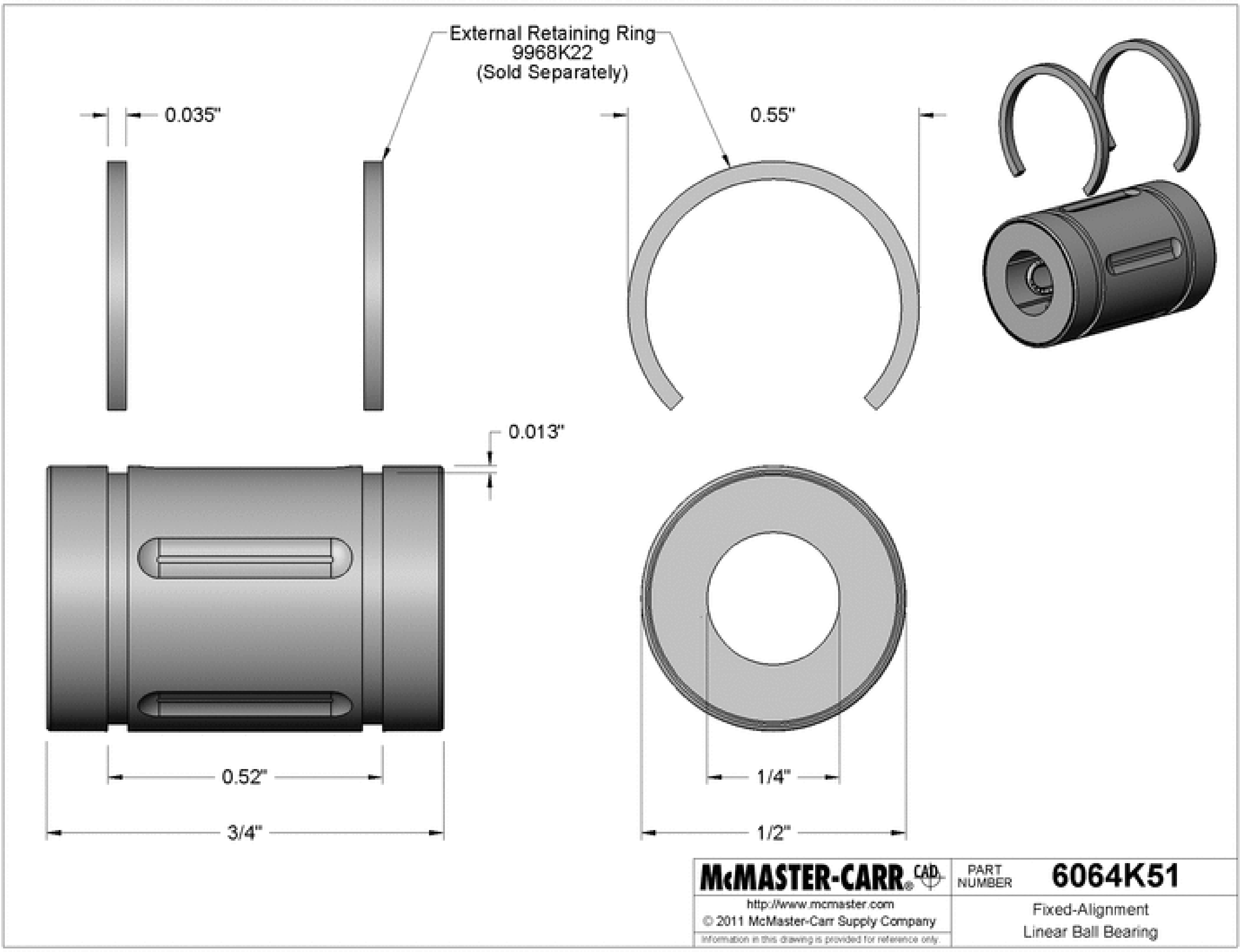
## Steel Bearing with Steel Ball, for 1/4" Shaft Diameter

In stock
$23.88 Each
6064K51




| | |
|---|---|
| Linear Bearing Component | Bearing |
| For Shaft Type | Round |
| Bearing Type | Ball |
| Alignment Style | Fixed |
| With Retaining Ring Grooves | Yes |
| For Shaft Diameter | 1/4" |
| ID | 0.250" |
| ID Tolerance | -0.0005" to 0" |
| Overall Length | 3/4" |
| For Housing ID | 0.500" - 0.501" |
| OD | 1/2" |
| OD Tolerance | 0" to 0.0005" |
| Retaining Ring Grooves Center-to-Center | 0.52" |
| With End Seals | No |
| Material | |
| Bearing | Steel |
| Ball | Steel |
| Dynamic Load Capacity | 15 lbs. |
| Static Load Capacity | Not Rated |
| Dynamic Load Capacity at Maximum Temperature | 15 lbs. @ 500° F |
| Lubrication | Required |
| Minimum Temperature | Not Rated |
| Maximum Temperature | 500° F |
| For Shaft Material | Steel, Stainless Steel |
| For Minimum Shaft Hardness | |
| Steel | Rockwell C56 |
| Stainless Steel | Rockwell C56 |
| Related Product | External Retaining Rings |

The choice for low-friction motion in high-temperature environments. Bearings are for use with round end-supported shafts and have a fixed-alignment design for applications where shaft misalignment is unlikely. To install, slide bearings into a housing (not included) and secure with two retaining rings (sold individually).

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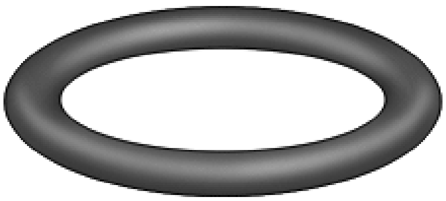

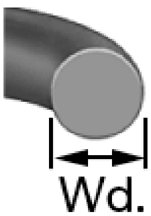

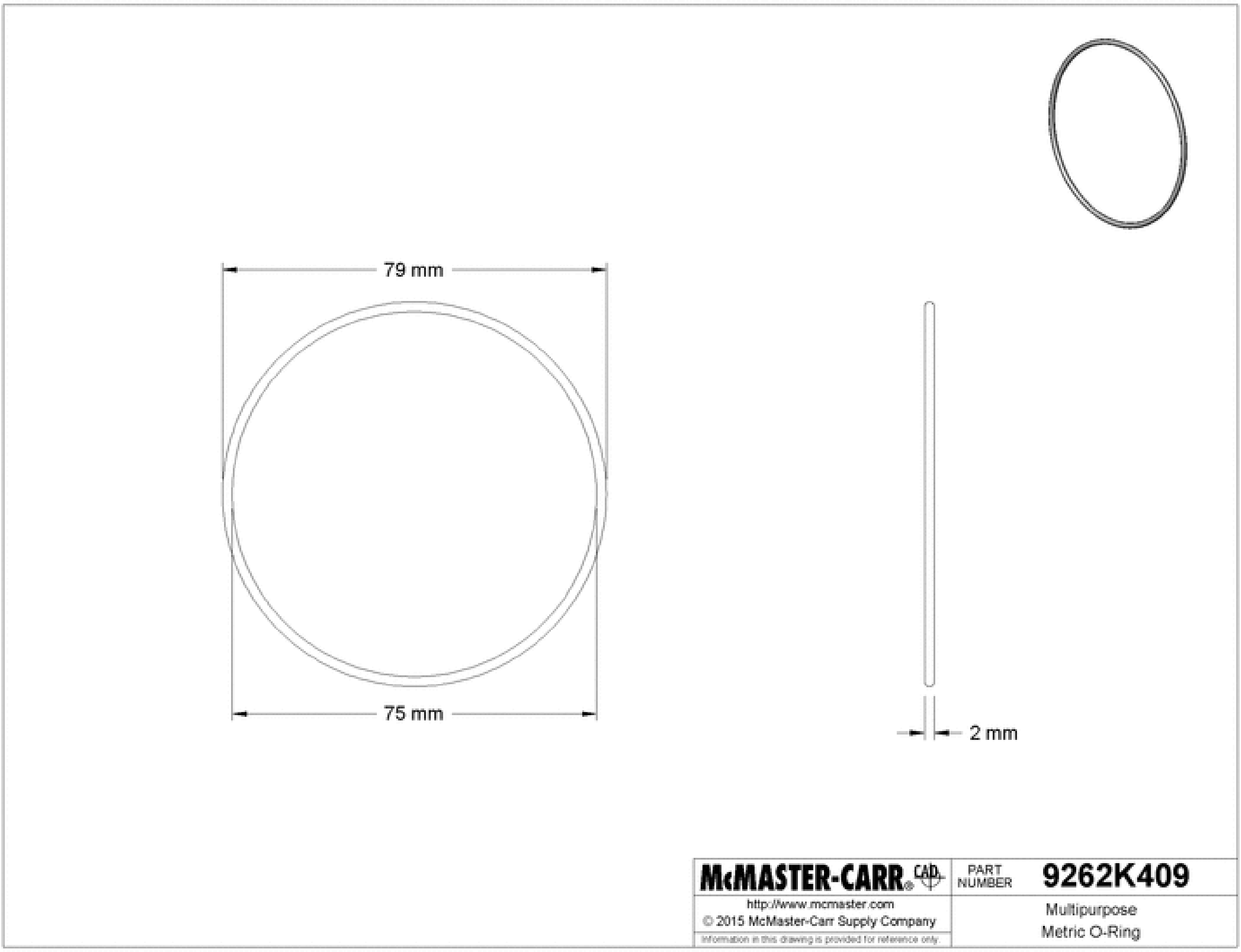


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Text 75930

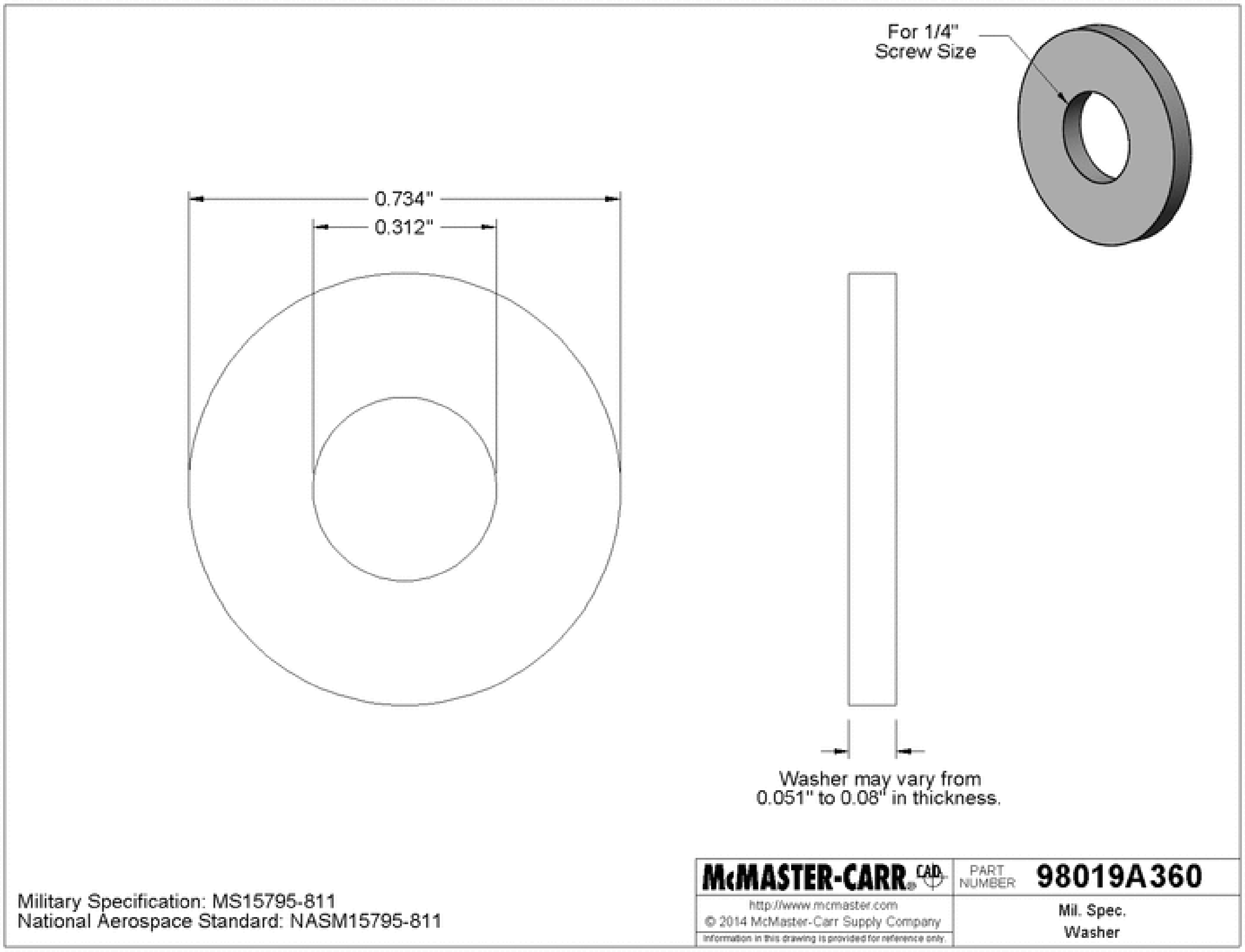
# Mil. Spec. 300 Series Stainless Steel Flat Washer

## Passivated, 1/4" Screw Size, MS/NASM-15795-811

In stock
$6.75 per pack of 25
98019A360

| | |
|---|---|
| Material | 300 Series Passivated Stainless Steel |
| Screw Size | 1/4" |
| Specifications Met | MS 15795/NASM 15795 |
| Dash No. | 811 |
| ID | 0.312" |
| OD | 0.734" |
| Thickness | |
| Minimum | 0.051" |
| Maximum | 0.080" |
| RoHS | Compliant |

Washers meet strict military standards to ensure performance and reliability.

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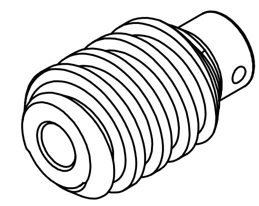
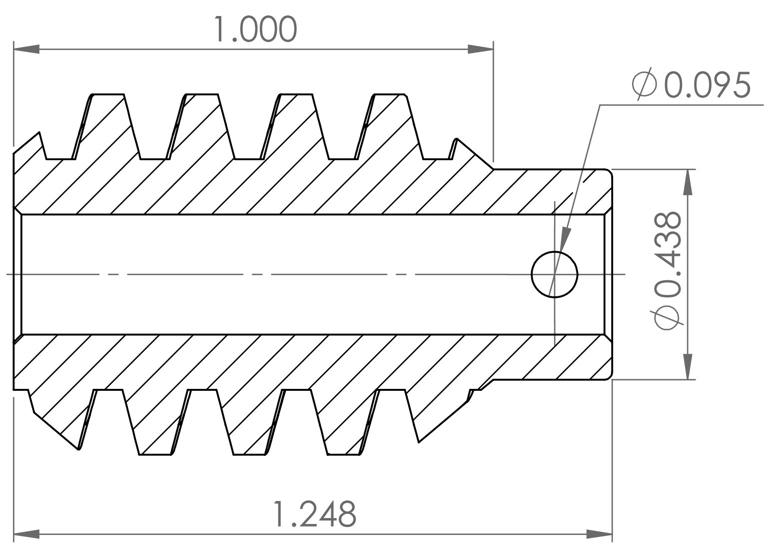
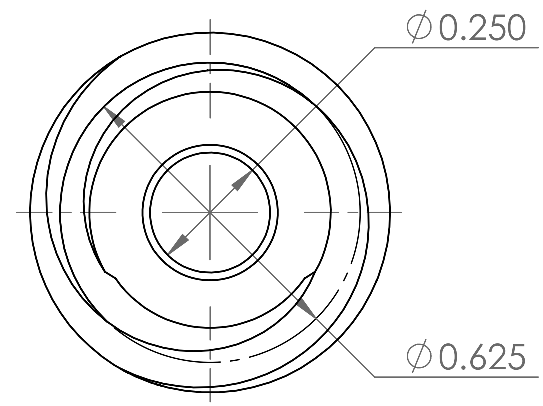





SIDE
(BREAKOUT)




FRONT

| | | |
|---|---|---|
| ITEM NAME:<br>Boston Gear LVHB1 Worm Gear, 14.5 Degree Pressure Angle, 0.250" Bore, 16 Pitch, .625 PD, RH | ASIN :<br>B004N6356I | MATERIAL :<br>Steel |
| | BRAND :<br>Boston Gear | # UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES<br># INFORMATION IN THIS DRAWING IS PROVIDED FOR REFERENCE ONLY |

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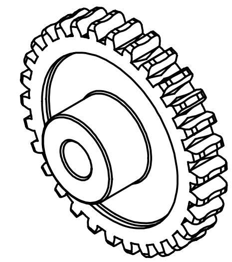
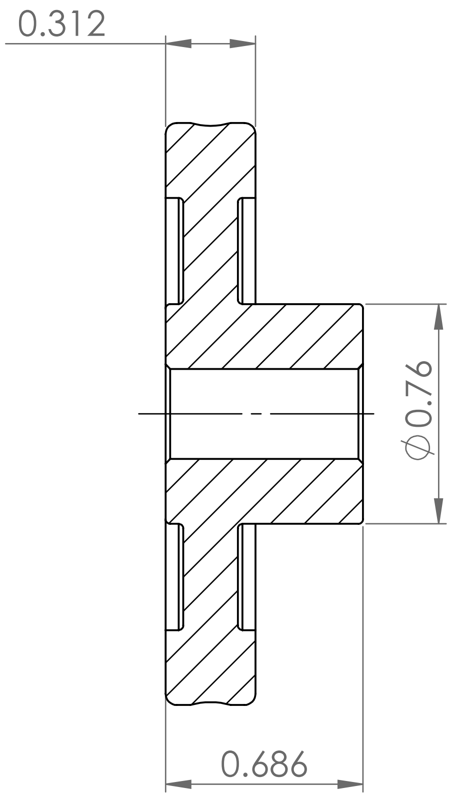
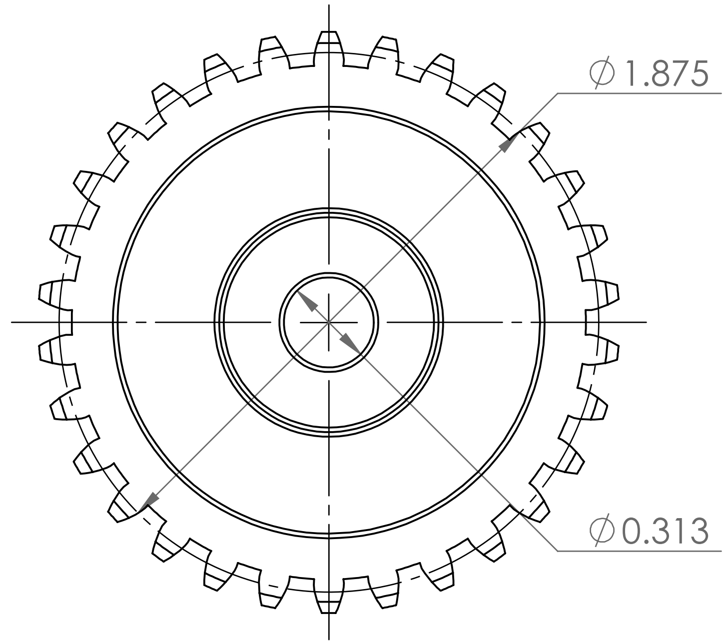
B004N84ZZQ






SIDE
(BREAKOUT)




FRONT

ITEM NAME:

Boston Gear G1043 Worm Gear, Web, 14.5 PA Pressure Angle, 0.313" Bore, 30:1 Ratio, 30 TEETH, RH

ASIN :

B004N84ZZQ

MATERIAL :

Bronze

BRAND :

Boston Gear

# UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES

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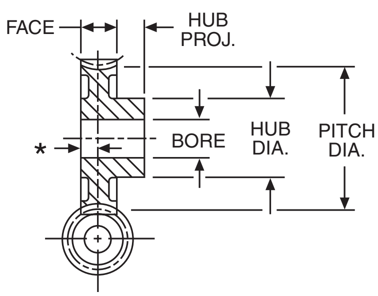
# WORMS AND WORM GEARS

## *16 DIAMETRAL PITCH*
## *BRONZE AND CAST IRON WORM GEARS*
## *STEEL WORMS – UNHARDENED AND HARDENED*

***PRESSURE ANGLE – SINGLE THREAD 14½°***
***DOUBLE THREAD 14½°***
***QUAD THREAD 20°***

RATIO = Gear Teeth ÷ Worm Threads
All Worms and Worm Gears stocked RIGHT HAND ONLY.




**ALL DIMENSIONS IN INCHES**
**ORDER BY CATALOG NUMBER OR ITEM CODE**

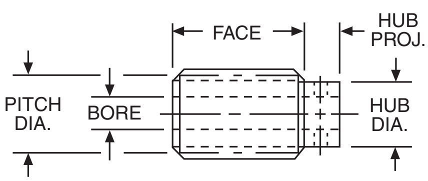
| **16 DIAMETRAL PITCH** | | | | | | **WORM GEARS** | | | | | **FACE = .313" *CENTER LINE WORM TO FLUSH END = .156"** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | **Hub** | | **Style** | **SINGLE Thread** | | **DOUBLE Thread** | | **QUAD Thread** | |
| **No. of Teeth** | **Pitch Dia.** | **Bore** | **Dia.** | **Proj.** | **See Page 150** | **Catalog Number** | **Item Code** | **Catalog Number** | **Item Code** | **Catalog Number** | **Item Code** |
| | | | | | | **BRONZE** | | | | | |
| 20 | 1.250 | .250 | .62 | .31 | A | G1042 | 13612 | D1142 | 13700 | Q1342 | 13536 |
| 30 | 1.875 | .3125 | .75 | .38 | B | G1043 | 13614 | D1143 | 13702 | Q1343 | 13538 |
| 40 | 2.500 | | | | C | G1044 | 13616 | D1144 | 13704 | Q1344 | 13540 |
| 50 | 3.125 | .375 | .88 | .44 | D | G1045 | 13618 | D1145 | 13706 | Q1345 | 13542 |
| 60 | 3.750 | | | | | G1048 | 13620 | D1148 | 13708 | Q1348 | 13544 |
| 80 | 5.000 | | | | | G1046 | 13622 | – | – | Q1346 | 13546 |
| 100 | 6.250 | | 1.00 | .50 | | G1047 | 13624 | – | – | – | – |
| | | | | | | **CAST IRON** | | | | | |
| 20 | 1.250 | .250 | .62 | .31 | A | CG1042 | 63506 | CD1142 | 63513 | CQ1342 | 63518 |
| 30 | 1.875 | .3125 | .75 | .38 | B | CG1043 | 63507 | CD1143 | 63514 | CQ1343 | 63519 |
| 40 | 2.500 | | | | C | CG1044 | 63508 | CD1144 | 63515 | CQ1344 | 63520 |
| 50 | 3.125 | .375 | .88 | .44 | | CG1045 | 63509 | CD1145 | 63516 | CQ1345 | 63521 |
| 60 | 3.750 | | | | | CG1048 | 63510 | CD1148 | 63517 | CQ1348 | 63522 |
| 80 | 5.000 | | | | | CG1046 | 63511 | – | – | CQ1346 | 63523 |
| 100 | 6.250 | | 1.00 | .50 | | CG1047 | 63512 | – | – | – | – |

### *STANDARD TOLERANCES*

| DIMENSION | | TOLERANCE |
|---|---|---|
| BORE | All | ±.0005 |

### *WORM LEAD and LEAD ANGLE*

| | SINGLE | DOUBLE | QUAD |
|---|---|---|---|
| LEAD | .1963" | .3927" | .7854" |
| LEAD ANGLE | 5°43´ | 11°19´ | 21°48´ |




| **16 DIAMETRAL PITCH** | | | | | **WORMS FOR ABOVE GEARS** | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | **Hub** | | **SINGLE Thread** | | **DOUBLE Thread** | | **QUAD Thread** | |
| **Pitch Dia.** | **Face** | **Bore** | **Dia.** | **Proj.** | **Catalog Number** | **Item Code** | **Catalog Number** | **Item Code** | **Catalog Number** | **Item Code** |
| | | | | | **UNHARDENED – STEEL** | | | | | |
| .625 | 1.00 | .250 | .44 | .25 | LVHB-1 | 12926 | DVH-1 | 12862 | QVH-1 | 12940 |
| | | | | | **HARDENED – STEEL** | | | | | |
| .625 | 1.00 | .3125 | .44 | .25 | HLVH-1<br>GLVH-1 | 13032<br>12958 | HDVH-1<br>GDVH-1 | 13004<br>12950 | HQVH-1<br>GQVH-1 | 13058<br>13046 |

All worms have .0938 drilled hole in hub.
Hxxx worms have polished threads.
Gxxx worms have ground and polished threads.

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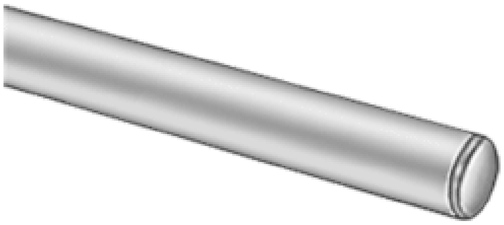
# Hardened Precision Shafts with Machinable Ends

## Annealed Both Ends, 1/4" Diameter, 10" Length

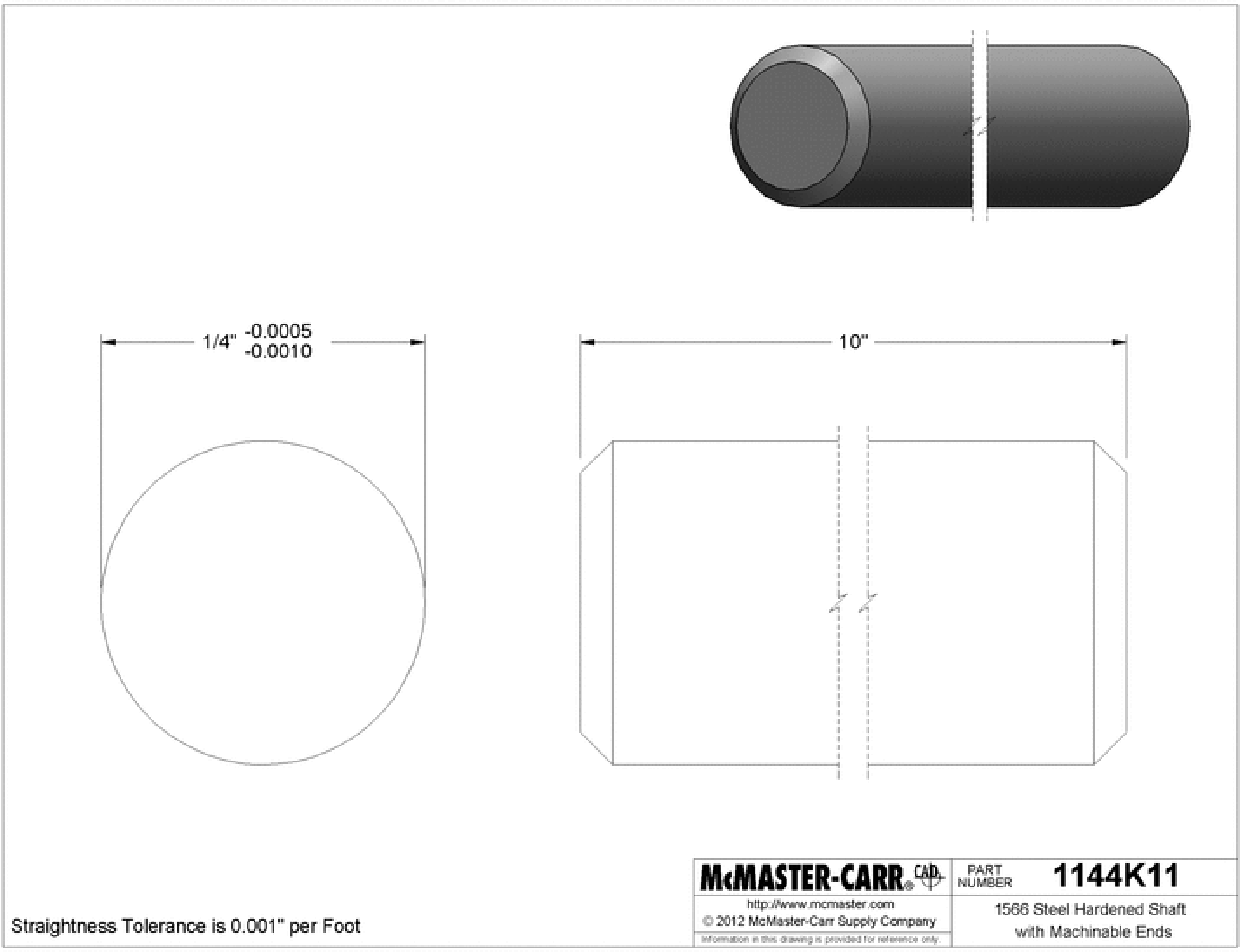
In stock
$24.15 Each
1144K11



| | |
|---|---|
| Material | 1566 Steel |
| Diameter | 1/4" |
| Diameter Tolerance | -0.001" to -0.0005" |
| Length | 10" |
| Straightness Tolerance | 0.001" per ft. |
| Case Hardness | Rockwell C60 |
| Minimum Hardness Depth | 0.027" |
| RoHS | Compliant |

Machining a custom end for mounting is easier because 2" of each end has been annealed (softened). Shafts are case hardened and precision ground with exacting diameter and straightness tolerances. The nonannealed portion has a 9 rms micron finish. Shafts are 1566 steel for high strength. Case hardness is Rockwell C60. Straightness tolerance is 0.001" per foot. Ends are beveled.

Note: The annealed (softened) ends are not intended for linear travel.




The information in this 3-D model is provided for reference only.

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# Stainless Steel Ball Bearing

## Double Shielded with Extended Inner Ring, Trade No. R4-2Z

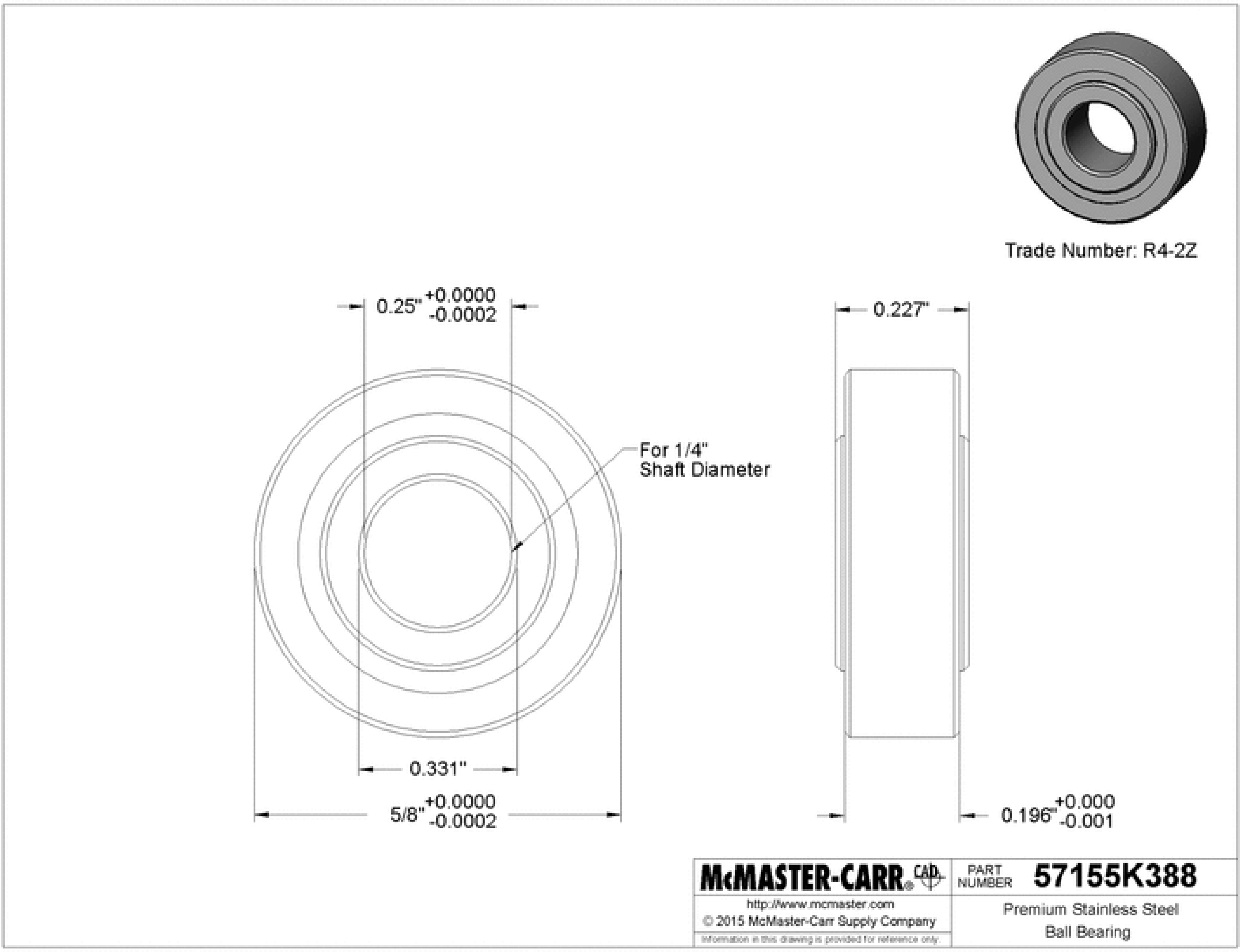
In stock
$4.78 Each
57155K388

| | |
|---|---|
| Bearing Type | Ball |
| For Load Direction | Radial |
| Ball Bearing Type | Extended Inner Ring |
| Construction | Single Row |
| Seal Type | Double Shielded |
| For Shaft Shape | Round |
| Trade No. | R4-2Z |
| For Shaft Diameter | 1/4" |
| ID | 0.25" |
| ID Tolerance | -0.0002" to 0" |
| OD | 5/8" |
| OD Tolerance | -0.0002" to 0" |
| Width | 0.196" |
| Width Tolerance | -0.001" to 0" |
| Overall Width | 0.227" |
| Inner Ring OD | 0.331" |
| Material | 440C Stainless Steel |
| Shield Material | Stainless Steel |
| Ball Material | Stainless Steel |
| Cage Material | Stainless Steel |
| Radial Load Capacity, lbs. | |
| Dynamic | 335 |
| Static | 125 |
| Maximum Speed | 45,000 rpm |
| Shaft Mount Type | Press Fit |
| Lubrication | Lubricated |
| Lubrication Method | Filled |
| Lubricant Type | Oil |
| Lubricant | Aeroshell #12 |
| Temperature Range | -60° to 250° F |
| ABEC Rating | ABEC-5 |
| Radial Clearance Trade No. | MC3 |
| Radial Clearance | 0" to 0" |
| RoHS | Compliant |

Made to tight tolerances, these 440C stainless steel bearings combine speed and accuracy with corrosion resistance.

Double-shielded bearings resist dust and other contaminants, but do not dissipate heat as efficiently as open bearings.

Bearings with extended inner ring provide additional shaft support.

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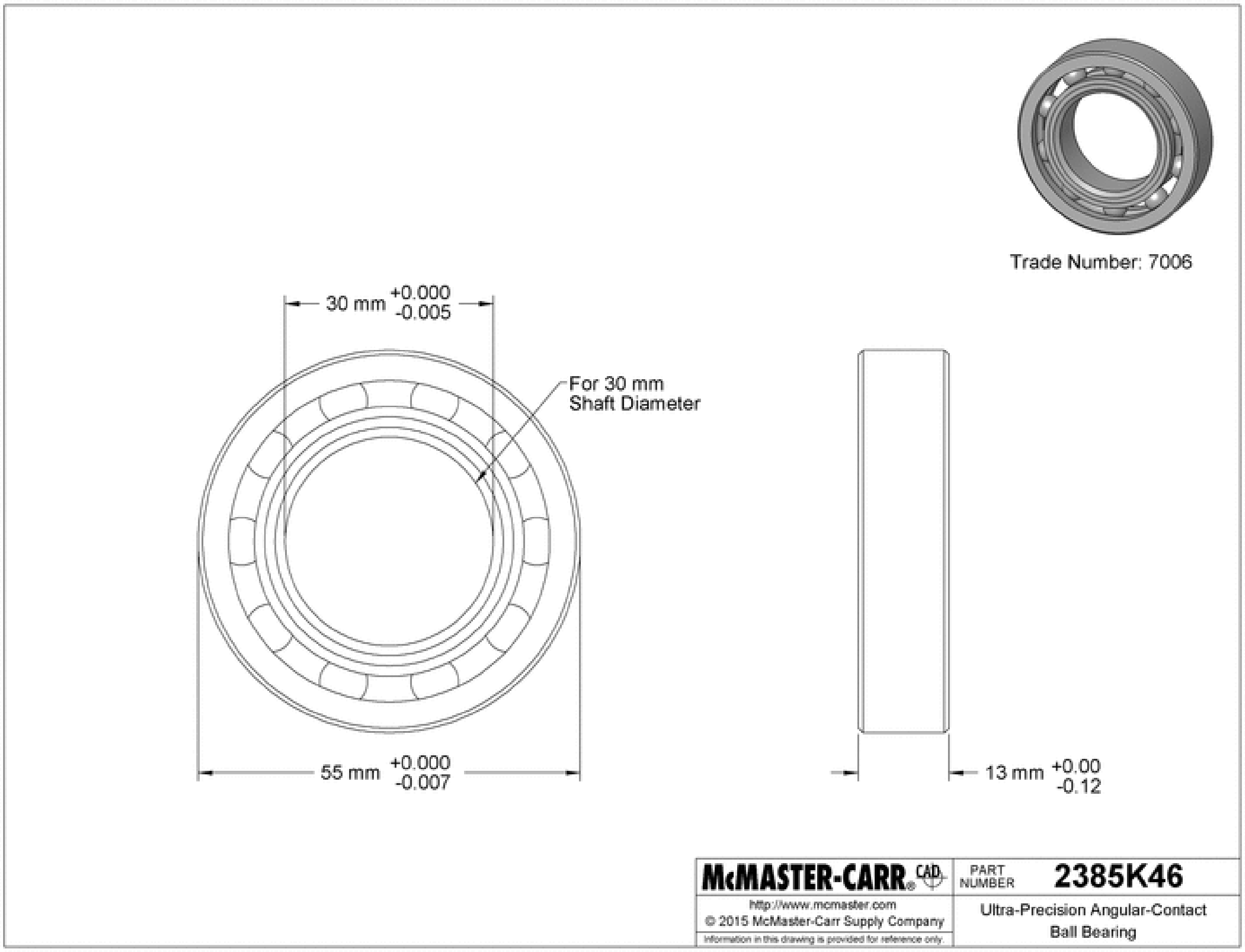
# Ultra-Precision Angular-Contact Ball Bearing

## Trade No. 7006

In stock
$272.75 Each
2385K46

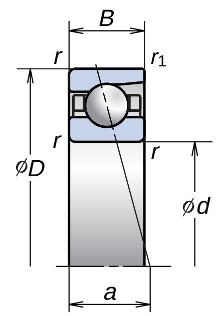
| | |
|---|---|
| Bearing Type | Ball |
| For Load Direction | Combined Radial and Thrust |
| Ball Bearing Type | Standard |
| Construction | Angular Contact, Single Row |
| Seal Type | Open |
| For Shaft Shape | Round |
| Trade No. | 7006 |
| For Shaft Diameter | 30 mm |
| ID | 30 mm |
| ID Tolerance | -0.005 to 0 mm |
| OD | 55 mm |
| OD Tolerance | -0.007 to 0 mm |
| Width | 13 mm |
| Width Tolerance | -0.12 to 0 mm |
| Material | 52100 Steel |
| Ball Material | Steel |
| Cage Material | Plastic |
| Radial Load Capacity, lbs. | |
| Dynamic | 3,220 |
| Static | 1,795 |
| Maximum Speed | 32,000 rpm |
| Shaft Mount Type | Press Fit |
| Lubrication | Required |
| Temperature Range | -30° to 140° F |
| ABEC Rating | ABEC-7 |
| Contact Angle | 15° |
| RoHS | Compliant |

Commonly found in machine tool spindles, these bearings are made to tight tolerances to perform in high-speed, precision applications. Mount them in pairs to handle thrust loads from both directions.




The information in this 3-D model is provided for reference only.

## High Precision Angular Contact Ball Bearings (Standard Series)




### 70 Series

Bore Diameter 10-75 mm

| Bearing Numbers | Boundary Dimensions (mm) | | | | | Basic Load Ratings (kN) | | Permissible Axial Load (1) (kN) | Factor $f_0$ | Effective Load Center (mm) $a$ | Mass (kg) (approx.) | Sealed Design | Limiting Speeds (2) (min⁻¹) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $d$ | $D$ | $B$ | $r$ (min.) | $r_1$ (min.) | $C_r$ (Dynamic) | $C_{0r}$ (Static) | | | | | | Grease | Oil |
| 7000C | 10 | 26 | 8 | 0.3 | 0.15 | 5.30 | 2.49 | 2.16 | 12.6 | 6.4 | 0.019 | – | 63 900 | 97 300 |
| 7000A5 | 10 | 26 | 8 | 0.3 | 0.15 | 5.15 | 2.41 | 2.48 | – | 8.2 | 0.019 | – | 55 600 | 83 400 |
| 7000A | 10 | 26 | 8 | 0.3 | 0.15 | 5.00 | 2.34 | 1.91 | – | 9.2 | 0.019 | – | 41 700 | 55 600 |
| 7001C | 12 | 28 | 8 | 0.3 | 0.15 | 5.80 | 2.90 | 2.40 | 13.2 | 6.7 | 0.021 | – | 57 500 | 87 500 |
| 7001A5 | 12 | 28 | 8 | 0.3 | 0.15 | 5.60 | 2.79 | 2.82 | – | 8.7 | 0.021 | – | 50 000 | 75 000 |
| 7001A | 12 | 28 | 8 | 0.3 | 0.15 | 5.40 | 2.71 | 2.13 | – | 9.8 | 0.021 | – | 37 500 | 50 000 |
| 7002C | 15 | 32 | 9 | 0.3 | 0.15 | 6.25 | 3.40 | 2.63 | 14.1 | 7.6 | 0.030 | – | 49 000 | 74 500 |
| 7002A5 | 15 | 32 | 9 | 0.3 | 0.15 | 5.95 | 3.25 | 3.05 | – | 10.0 | 0.030 | – | 42 600 | 63 900 |
| 7002A | 15 | 32 | 9 | 0.3 | 0.15 | 5.80 | 3.15 | 2.36 | – | 11.3 | 0.030 | – | 32 000 | 42 600 |
| 7003C | 17 | 35 | 10 | 0.3 | 0.15 | 6.60 | 3.80 | 2.85 | 14.5 | 8.5 | 0.039 | – | 44 300 | 67 400 |
| 7003A5 | 17 | 35 | 10 | 0.3 | 0.15 | 6.30 | 3.65 | 3.35 | – | 11.1 | 0.040 | – | 38 500 | 57 700 |
| 7003A | 17 | 35 | 10 | 0.3 | 0.15 | 6.10 | 3.50 | 2.59 | – | 12.5 | 0.040 | – | 28 900 | 38 500 |
| 7004C | 20 | 42 | 12 | 0.6 | 0.3 | 11.1 | 6.55 | 4.80 | 14.0 | 10.1 | 0.067 | – | 37 100 | 56 500 |
| 7004A5 | 20 | 42 | 12 | 0.6 | 0.3 | 10.6 | 6.25 | 5.45 | – | 13.2 | 0.067 | – | 32 300 | 48 400 |
| 7004A | 20 | 42 | 12 | 0.6 | 0.3 | 10.3 | 6.10 | 4.20 | – | 14.9 | 0.068 | – | 24 200 | 32 300 |
| 7005C | 25 | 47 | 12 | 0.6 | 0.3 | 11.7 | 7.40 | 5.20 | 14.7 | 10.8 | 0.078 | – | 32 000 | 48 700 |
| 7005A5 | 25 | 47 | 12 | 0.6 | 0.3 | 11.1 | 7.10 | 5.95 | – | 14.4 | 0.077 | – | 27 800 | 41 700 |
| 7005A | 25 | 47 | 12 | 0.6 | 0.3 | 10.7 | 6.85 | 4.55 | – | 16.4 | 0.079 | – | 20 900 | 27 800 |
| 7006C | 30 | 55 | 13 | 1.0 | 0.6 | 15.1 | 10.3 | 6.85 | 14.9 | 12.2 | 0.114 | ○ | 27 100 | 41 200 |
| 7006A5 | 30 | 55 | 13 | 1.0 | 0.6 | 14.4 | 9.80 | 8.05 | – | 16.4 | 0.114 | ○ | 23 600 | 35 300 |
| 7006A | 30 | 55 | 13 | 1.0 | 0.6 | 13.9 | 9.45 | 6.20 | – | 18.8 | 0.116 | ○ | 17 700 | 23 600 |
| 7007C | 35 | 62 | 14 | 1.0 | 0.6 | 19.1 | 13.7 | 9.35 | 15.0 | 13.5 | 0.151 | ○ | 23 800 | 36 100 |
| 7007A5 | 35 | 62 | 14 | 1.0 | 0.6 | 18.2 | 13.0 | 11.4 | – | 18.3 | 0.151 | ○ | 20 700 | 31 000 |
| 7007A | 35 | 62 | 14 | 1.0 | 0.6 | 17.5 | 12.6 | 8.75 | – | 21.0 | 0.153 | ○ | 15 500 | 20 700 |

(1) For permissible axial load, please refer to Page 147.
(2) For application of limiting speeds, please refer to Page 170.
When a ceramic ball is used, limiting speed value will be 1.25 times the value of steel ball.

Note: Bearing numbers with a "C" suffix: nominal contact angle 15°
Bearing numbers with an "A5" suffix: nominal contact angle 25°
Bearing numbers with an "A" suffix: nominal contact angle 30°

For additional information: Page No.
- Dynamic equivalent load ······ 139
- Static equivalent load ······ 146
- Preload and rigidity ······ 152
- Abutment and fillet dimensions ······ 186
- Nozzle position ······ 192
- Quantity of packed grease ······ 175

### 70 Series (continued)

| Bearing Numbers | Boundary Dimensions (mm) | | | | | Basic Load Ratings (kN) | | Permissible Axial Load (1) (kN) | Factor $f_0$ | Effective Load Center (mm) $a$ | Mass (kg) (approx.) | Sealed Design | Limiting Speeds (2) (min⁻¹) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $d$ | $D$ | $B$ | $r$ (min.) | $r_1$ (min.) | $C_r$ (Dynamic) | $C_{0r}$ (Static) | | | | | | Grease | Oil |
| 7008C | 40 | 68 | 15 | 1.0 | 0.6 | 20.6 | 15.9 | 10.6 | 15.4 | 14.7 | 0.189 | ○ | 21 300 | 32 500 |
| 7008A5 | 40 | 68 | 15 | 1.0 | 0.6 | 19.5 | 15.1 | 12.0 | – | 20.1 | 0.188 | ○ | 18 600 | 27 800 |
| 7008A | 40 | 68 | 15 | 1.0 | 0.6 | 18.8 | 14.6 | 9.15 | – | 23.1 | 0.191 | ○ | 13 900 | 18 600 |
| 7009C | 45 | 75 | 16 | 1.0 | 0.6 | 24.4 | 19.3 | 12.4 | 15.4 | 16.0 | 0.238 | ○ | 19 200 | 29 200 |
| 7009A5 | 45 | 75 | 16 | 1.0 | 0.6 | 23.1 | 18.3 | 14.5 | – | 22.0 | 0.250 | ○ | 16 700 | 25 000 |
| 7009A | 45 | 75 | 16 | 1.0 | 0.6 | 22.3 | 17.7 | 11.1 | – | 25.3 | 0.241 | ○ | 12 500 | 16 700 |
| 7010C | 50 | 80 | 16 | 1.0 | 0.6 | 26.0 | 21.9 | 13.9 | 15.7 | 16.7 | 0.259 | ○ | 17 700 | 27 000 |
| 7010A5 | 50 | 80 | 16 | 1.0 | 0.6 | 24.6 | 20.8 | 16.2 | – | 23.2 | 0.270 | ○ | 15 400 | 23 100 |
| 7010A | 50 | 80 | 16 | 1.0 | 0.6 | 23.7 | 20.1 | 12.5 | – | 26.8 | 0.262 | ○ | 11 600 | 15 400 |
| 7011C | 55 | 90 | 18 | 1.1 | 0.6 | 34.0 | 28.6 | 18.9 | 15.5 | 18.7 | 0.380 | ○ | 15 900 | 24 200 |
| 7011A5 | 55 | 90 | 18 | 1.1 | 0.6 | 32.5 | 27.2 | 21.8 | – | 25.9 | 0.383 | ○ | 13 800 | 20 700 |
| 7011A | 55 | 90 | 18 | 1.1 | 0.6 | 31.0 | 26.3 | 16.6 | – | 29.9 | 0.385 | ○ | 10 400 | 13 800 |
| 7012C | 60 | 95 | 18 | 1.1 | 0.6 | 35.0 | 30.5 | 19.9 | 15.7 | 19.4 | 0.405 | ○ | 14 900 | 22 600 |
| 7012A5 | 60 | 95 | 18 | 1.1 | 0.6 | 33.0 | 29.1 | 23.0 | – | 27.1 | 0.408 | ○ | 13 000 | 19 400 |
| 7012A | 60 | 95 | 18 | 1.1 | 0.6 | 32.0 | 28.1 | 17.6 | – | 31.4 | 0.410 | ○ | 9 700 | 13 000 |
| 7013C | 65 | 100 | 18 | 1.1 | 0.6 | 37.0 | 34.5 | 22.0 | 15.9 | 20.0 | 0.435 | ○ | 14 000 | 21 300 |
| 7013A5 | 65 | 100 | 18 | 1.1 | 0.6 | 35.0 | 32.5 | 25.4 | – | 28.2 | 0.455 | ○ | 12 200 | 18 200 |
| 7013A | 65 | 100 | 18 | 1.1 | 0.6 | 33.5 | 31.5 | 19.5 | – | 32.8 | 0.441 | ○ | 9 100 | 12 200 |
| 7014C | 70 | 110 | 20 | 1.1 | 0.6 | 47.0 | 43.0 | 26.8 | 15.7 | 22.1 | 0.606 | ○ | 12 800 | 19 500 |
| 7014A5 | 70 | 110 | 20 | 1.1 | 0.6 | 44.5 | 41.0 | 32.0 | – | 31.0 | 0.625 | ○ | 11 200 | 16 700 |
| 7014A | 70 | 110 | 20 | 1.1 | 0.6 | 42.5 | 39.5 | 24.6 | – | 36.0 | 0.613 | ○ | 8 400 | 11 200 |
| 7015C | 75 | 115 | 20 | 1.1 | 0.6 | 48.0 | 45.5 | 28.1 | 15.9 | 22.7 | 0.643 | ○ | 12 200 | 18 500 |
| 7015A5 | 75 | 115 | 20 | 1.1 | 0.6 | 45.5 | 43.5 | 33.5 | – | 32.1 | 0.652 | ○ | 10 600 | 15 800 |
| 7015A | 75 | 115 | 20 | 1.1 | 0.6 | 43.5 | 41.5 | 25.9 | – | 37.4 | 0.650 | ○ | 7 900 | 10 600 |

(1) For permissible axial load, please refer to Page 147.
(2) For application of limiting speeds, please refer to Page 170.
When a ceramic ball is used, limiting speed value will be 1.25 times the value of steel ball.

Note: Bearing numbers with a "C" suffix: nominal contact angle 15°
Bearing numbers with an "A5" suffix: nominal contact angle 25°
Bearing numbers with an "A" suffix: nominal contact angle 30°

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# Servomotor Flexible Shaft Coupling

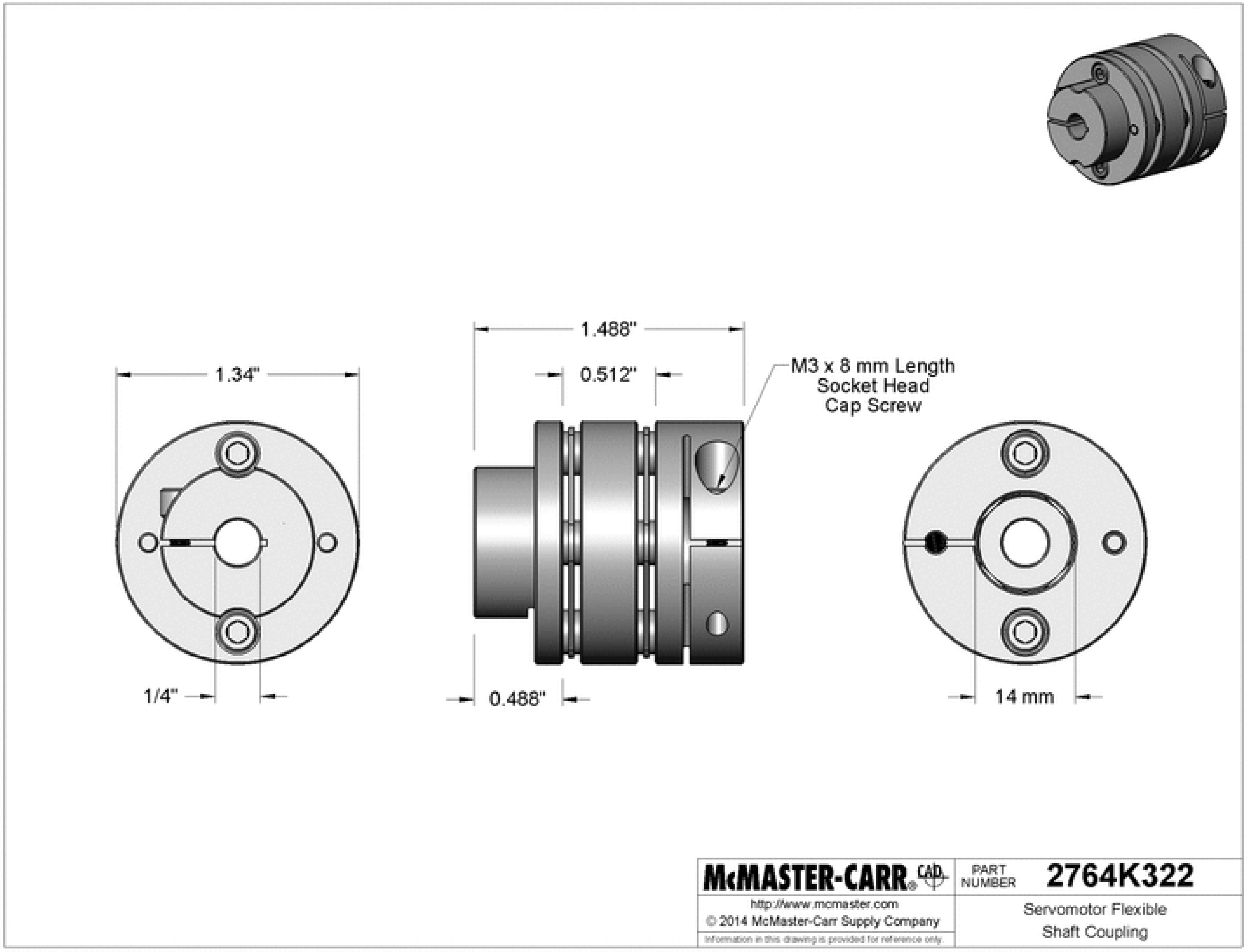
## 1/4" x 14mm Shaft Diameter, 1.488" Length, 1.340" OD

In stock
$78.69 Each
2764K322


| | |
|---|---|
| For Shaft Dia. (A) | 1/4" |
| For Shaft Dia. (B) | 14 mm |
| Length | 1.488" |
| OD | 1.34" |
| Maximum rpm | 10,000 |
| Maximum Torque | 35 in.-lbs. |
| Maximum Misalignment | |
| Parallel | 0.007" |
| Angular | 1° |
| Axial | 0.016" |

Able to handle high twisting forces as well as misalignment with zero backlash (no play), these are the choice for high-performance servomotor applications. Couplings are aluminum and Type 304 stainless steel for corrosion resistance. They fasten onto your shafts for superior holding power without damaging them. Tighten the two socket-head cap screws to secure.

For Shaft Diameter (A): 1/4", For Shaft Diameter (B): 14 mm

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# Servomotor Flexible Shaft Coupling

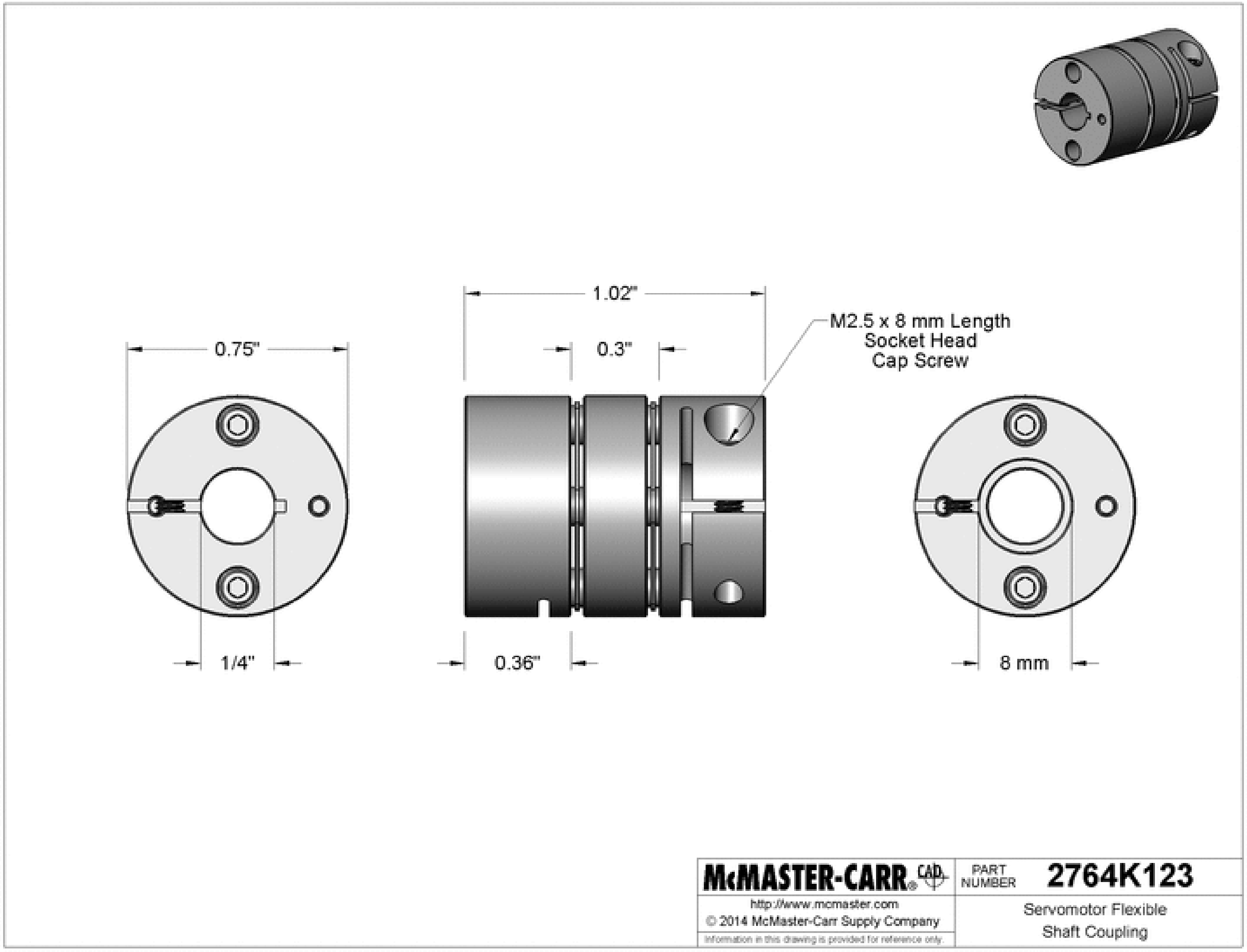
## 1/4" x 8mm Shaft Diameter, 1.02" Length, .75 OD

In stock
$60.54 Each
2764K123


| | |
|---|---|
| For Shaft Dia. (A) | 1/4" |
| For Shaft Dia. (B) | 8 mm |
| Length | 1.02" |
| OD | 0.75" |
| Maximum rpm | 10,000 |
| Maximum Torque | 7 in.-lbs. |
| Maximum Misalignment | |
| Parallel | 0.004" |
| Angular | 1° |
| Axial | 0.008" |

Able to handle high twisting forces as well as misalignment with zero backlash (no play), these are the choice for high-performance servomotor applications. Couplings are aluminum and Type 304 stainless steel for corrosion resistance. They fasten onto your shafts for superior holding power without damaging them. Tighten the two socket-head cap screws to secure.

For Shaft Diameter (A): 1/4", For Shaft Diameter (B): 8 mm




The information in this 3-D model is provided for reference only.

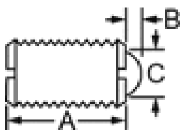
McMASTER-CARR. OVER 555,000 PRODUCTS

(562) 692-5911
(562) 695-2323 (fax)
la.sales@mcmaster.com
Text 75930

# Ball-Nose Spring Plunger with Stainless Steel Ball

## Steel Body, 1/4"-20 Thread, 4-12 lb. Nose Force

$3.62 Each
3408A75




| | |
|---|---|
| Thread Size | 1/4"-20 |
| Body Length (A) | 0.531" |
| Nose Length (B) | 0.035" |
| Nose Diameter (C) | 0.125" |
| Nose Force, lbs. | |
| Extended | 4 |
| Compressed | 12 |
| Thread-Locking Element | With Thread-Locking Element, Without Thread-Locking Element |
| Additional Specifications | Steel Body<br>Stainless Steel Ball<br>Inch |
| RoHS | Compliant |

The ball nose in these spring plungers rolls to minimize wear on the workpiece. They are compact for high-strength. Install with a screwdriver or plunger wrench (sold separately).

Stainless steel balls are 440C, except Type 316 stainless steel plungers have Type 316 stainless steel balls.

Maximum temperature is 250° F for steel and 18-8 stainless steel plungers with stainless steel ball and 170° F for all others.

**To Order:** Please specify with or without a thread-locking element. When threading into aluminum and other soft materials, select a plunger without the thread-locking element.

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# Zinc-Plated Alloy Steel Socket Head Cap Screw

## 1/4"-20 Thread, 1/2" Length

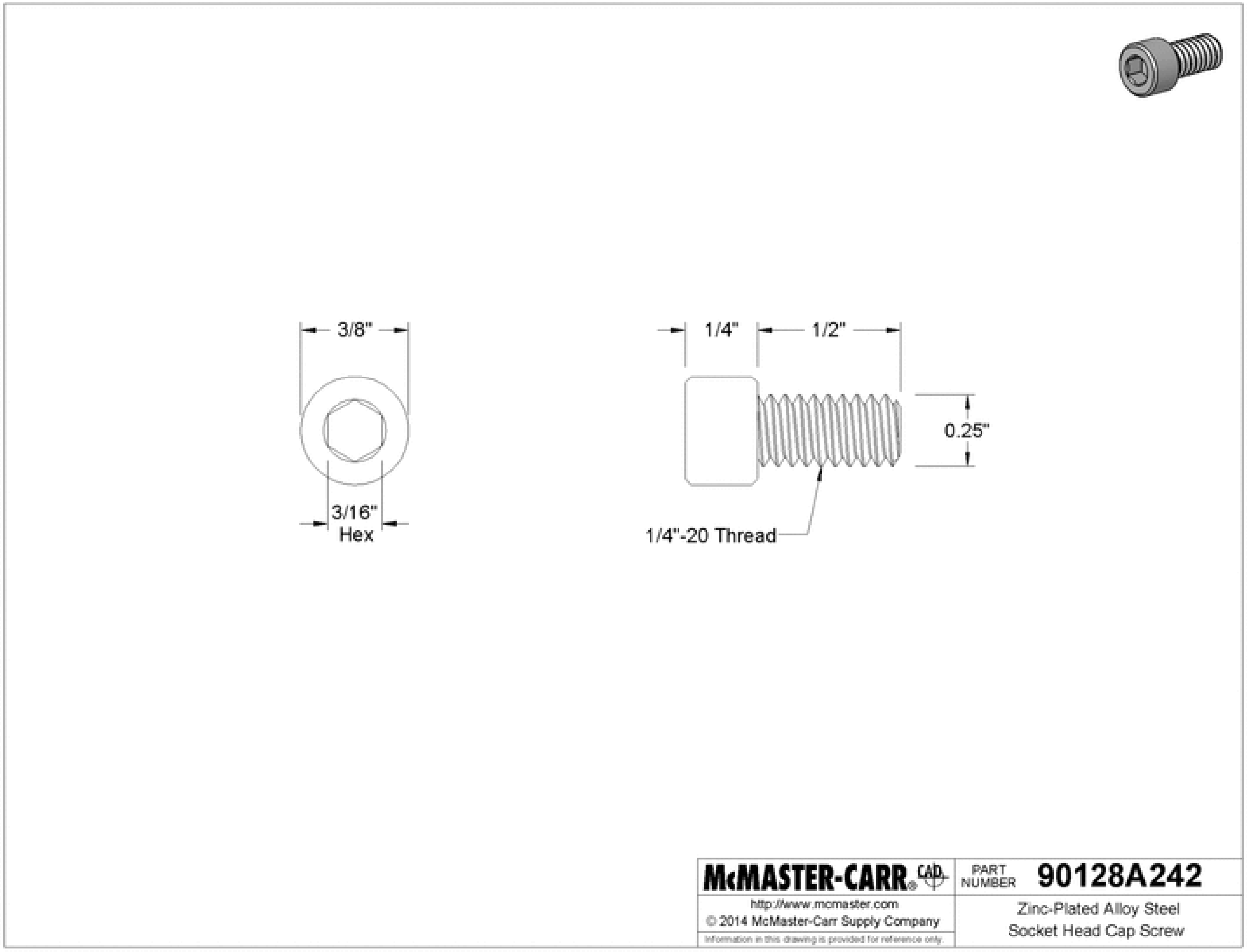
In stock
$9.09 per pack of 50
90128A242

| | |
|---|---|
| Thread Size | 1/4"-20 |
| Length | 1/2" |
| Thread Length | Full |
| Additional Specifications | Zinc-Plated Alloy Steel |

The standard among high-strength fasteners, these screws are stronger than Grade 8 steel screws. They have a minimum tensile strength of 170,000 psi. and a minimum Rockwell hardness of C37. Length is measured from under the head.

Inch screws have a Class 3A thread fit. They meet ASTM A574.

Zinc Plated—Screws have a protective coating that adds corrosion resistance for outdoor use.

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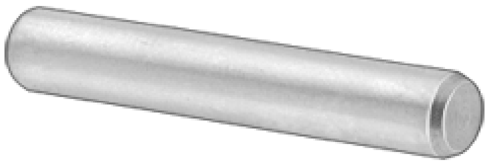
# Corrosion Resistant Dowel Pin

## Type 316 Stainless Steel, 3/16" Diameter, 1-1/8" Length

In stock
$11.25 per pack of 5
97395A474

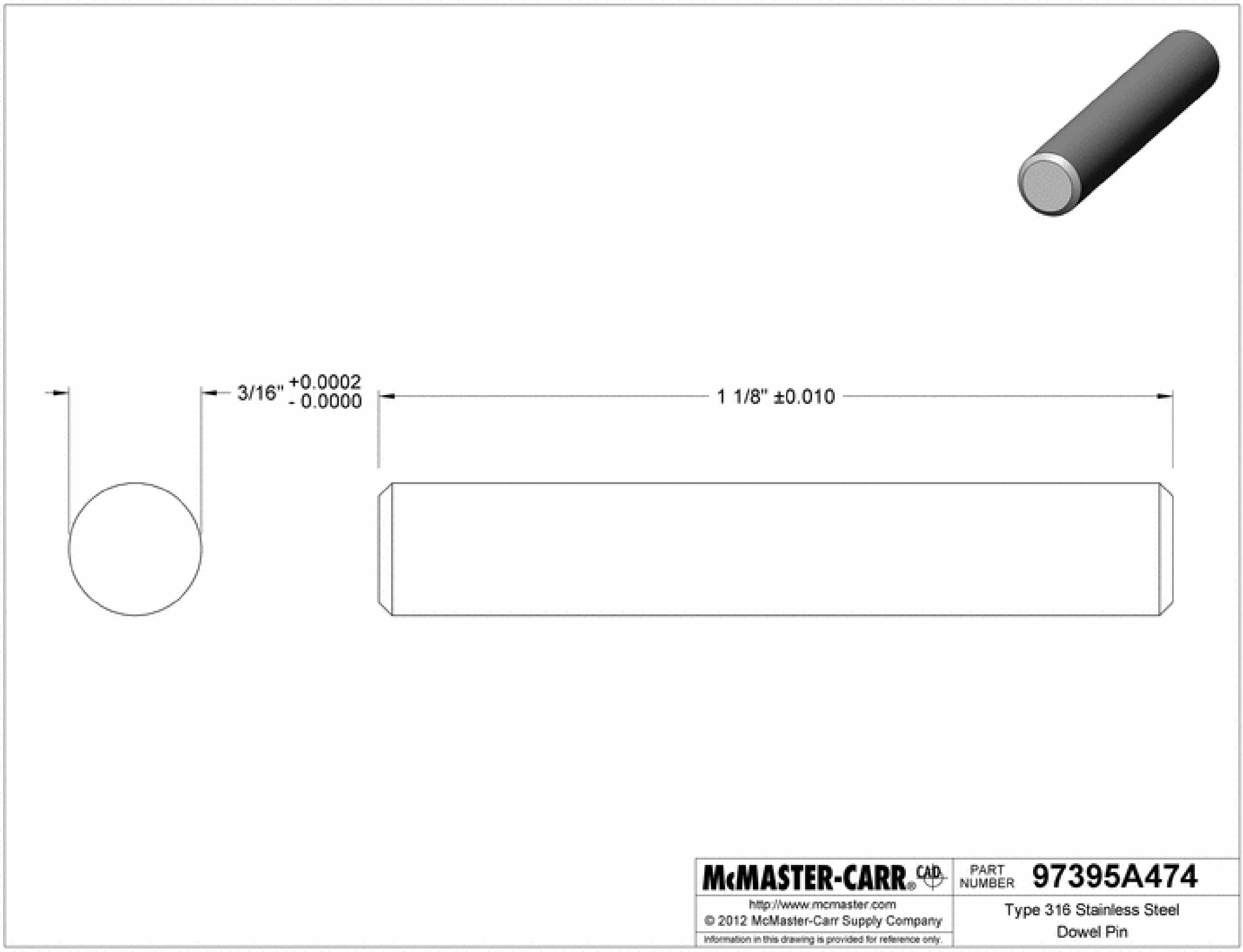


| | |
|---|---|
| Length | 1 1/8" |
| Additional Specifications | Type 316 Stainless Steel<br>3/16" Dia. |
| RoHS | Compliant |

Pins are commonly used as pivots, hinges, shafts, jigs, and fixtures to locate or hold parts. They are precision ground (unless noted) for more accurate alignment, and slightly oversized for a tight fit. To aid insertion, ends are beveled or rounded (unless noted). For a tight fit, your hole should be equal to or slightly smaller than the diameter shown. Breaking strength and Rockwell hardness are not rated (unless noted). Materials that have a breaking strength are measured as double shear, which is the force required to break a pin into three pieces.

Type 316 Stainless Steel—Our most corrosion resistant dowel pins. May be magnetic. Diameter tolerance is +0.0002".

Not rated for breaking strength.

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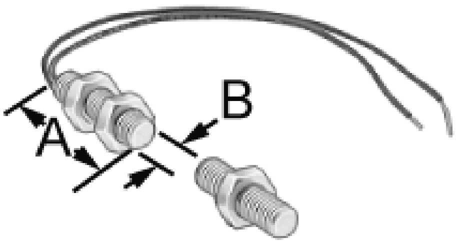
# Magnetically Actuated Switch

## Mini Cylindrical, SPDT, 150V DC

In stock
$46.11 Each
65985K22




| | |
|---|---|
| Amp Rating | 0.8 amps @ 24V DC |
| Maximum Voltage | 150 DC |
| Sensing Distance | 0.2" |
| (A) | 1.25" |
| (B) | 0.31" |
| Housing | Stainless Steel |
| Mounting Information | 5/16"-24 Threads |
| Additional Specifications | Standard<br>DC Rated—Switch One Circuit from Off to On or On to Off (SPDT)<br>Mini Cylindrical |
| RoHS | Compliant |
| Shipping | Regulated by the U.S. Department of Transportation |

With no moving parts to wear out, these switches provide long-term, trouble-free service. When the magnet comes within the sensing distance of the switch, the switch actuates. When the magnet moves away, the switch resets. The switch is usually mounted in a stationary location such as a door frame while the magnet is mounted on a movable object such as a door. Switches with mounting slots do not include hardware; switches with mounting threads include two hex nuts.

Standard switches are designed for proximity-sensing applications such as detecting when a door or window is open. Magnet included. They are not for use on safety-guard doors found on machinery. DC-rated switches have two wire leads except SPDT switches have three. UL and C-UL recognized.

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Text 75930

# Metric High-Precision Steel Needle-Roller Bearing

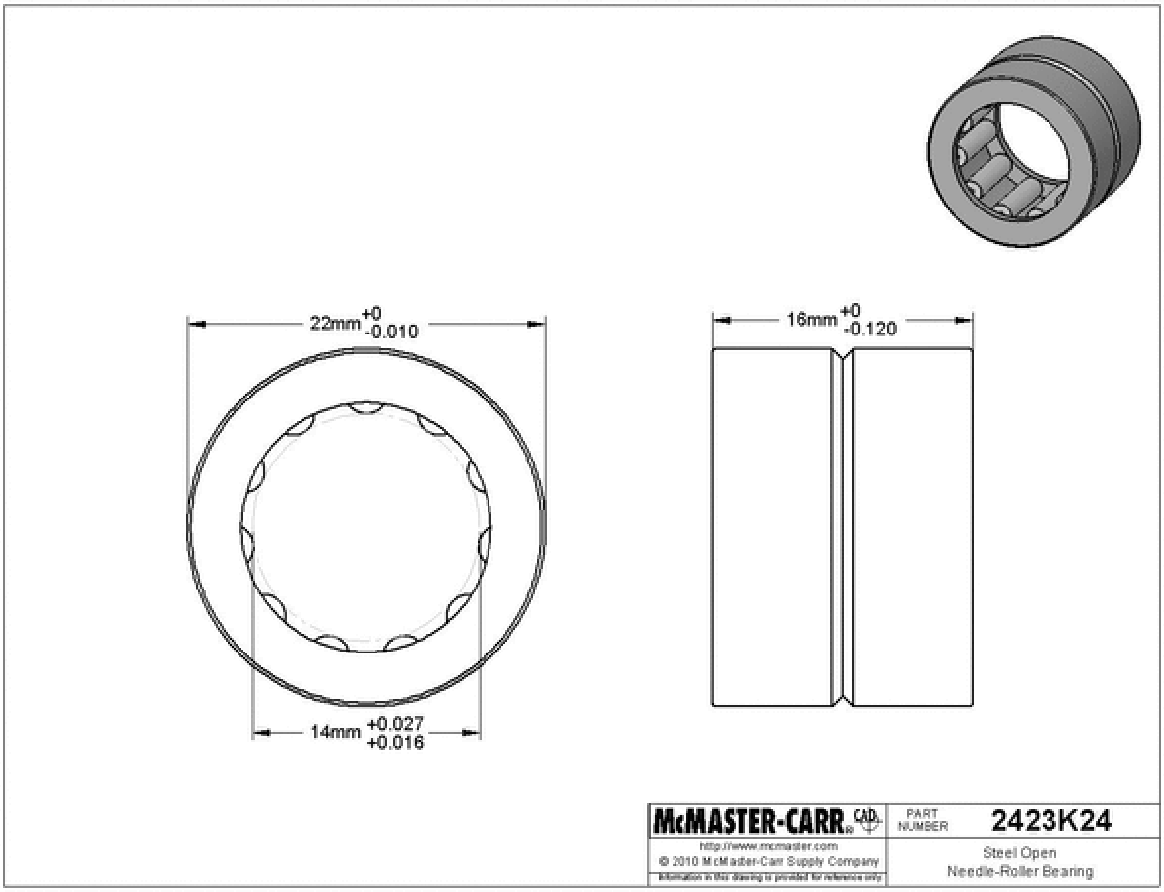
## for 14 mm Shaft Diameter, 22 mm Outside Diameter

In stock
$19.11 Each
2423K24

| | |
|---|---|
| Bearing Type | Roller |
| Roller Type | Needle |
| For Load Direction | Radial |
| Shaft Mount Type | Press Fit |
| Seal Type | Open |
| For Shaft Diameter | 14 mm |
| ID | 14.000 mm |
| ID Tolerance | 0.0160 to 0.0270 mm |
| OD | 22 mm |
| OD Tolerance | -0.0100 to 0.0000 mm |
| Width | 16 mm |
| Width Tolerance | -0.1200 to 0.0000 mm |
| Material | Steel |
| Lubrication | Required |
| Radial Load Capacity, lbs. | |
| Dynamic | 2,270 |
| Static | 2,600 |
| Maximum Speed | 24,000 rpm |
| Temperature Range | 0° to 250° F |
| For Shaft Smoothness | 4 microinches RA |
| Related Product | Optional Shaft Liners |

These precision-ground bearings give you higher load capacity, speed, and accuracy than standard needle-roller bearings. They're for use on hardened and ground shafts. If your shaft is not hardened and ground, use these bearings with liners (sold separately). Insert the liner into the bearing and press fit onto a shaft. All bearings have a lubrication groove and hole, except for the bearings for 8 and 10 mm shaft diameters.

Open dissipate heat more efficiently than double-sealed bearings.




The information in this 3-D model is provided for reference only.

# Appendix E

Drive Shaft Analysis

Driven Shaft Analysis

Inertia Analysis

Worm Gear A Analysis

Worm Gear B Analysis

Bearing Analysis

Bolts Analysis

*Drive Shaft for 5th Axis Rotary*
*This code solves for the drive shaft deflection in the 4th and 5th rotary stages to prevent backlash while preventing failure. This code also recommends a spring constant and calculates deflection from a given force. The shaft is 303 Stainless centerless ground*

*Geometric Parameters*

$D_{Shaft} = 0.25$ [in] *Shaft Diameter*

$L = 4$ [in] *Shaft Length*

*Material Properties*

$S_y = \textbf{YieldStress}\left[\text{Stainless}_{AISI302}, T = 62.33 \text{ [F]}\right]$

$S_{ut} = \textbf{UltimateStress}\left[\text{Stainless}_{AISI302}, T = 62.33 \text{ [F]}\right]$

$E = \textbf{YoungsModulus}\left[\text{Stainless}_{AISI302}, T = 62.33 \text{ [F]}\right]$

$a = 1.34$ *Ground Surface Finish (Table 6-2)*

$b = -0.085$ *Ground Surface Finish (Table 6-2)*

*Spring Parameters*

$\delta_{Shaft} = 0.01$ [in] *Ideal Shaft Deflection*

$\delta_{Spring} = 0.15$ [in] *Ideal Spring Deflection*

$Force = 4$ [lbf] *Amount of Force to apply to the Shaft*

*Other*

$T_s = 150$ [F] *Case Temperature as calculated in Gear Wear*

*Solve for Spring Constant*

$$I = \pi \cdot \frac{\left[\frac{D_{Shaft}}{2}\right]^4}{4}$$ *[in$^4$] Moment of Inertia for the Drive Shaft*

$$k = \delta_{Shaft} \cdot 3 \cdot E \cdot \frac{I}{\delta_{Spring} \cdot L^3}$$ *[lbf/in] Spring Constant for given deflections*

*Solve for Shaft Deflection Due to Assumed Force*

$$ShaftDeflection = Force \cdot \frac{L^3}{3 \cdot E \cdot I}$$ *[in] Resulting Shaft Deflection*

*Total Stress*

$M_z = Force \cdot L$ *[lbf*in] Moment in the z for the shaft*

$$\sigma_m = \frac{32}{\pi \cdot D_{Shaft}^3} \cdot M_z$$ *[psi] Maximum Bending Stress for a solid circular cross section (Equation 3-38)*

$$A_{Shaft} = \pi \cdot \left[ \frac{D_{Shaft}}{2} \right]^2$$ *[in²] Cross Sectional Area of the Shaft*

$$\tau = 4 \cdot \frac{Force}{3 \cdot A_{Shaft}}$$ *[psi] Transverse Shear Stress (Equation 3-31)*

$$\sigma_{Total} = \frac{\frac{\sigma_m}{2} + \sqrt{\left[\frac{\sigma_m}{2}\right]^2 + \tau^2}}{1000 \quad [psi/ksi]}$$ *[psi] Total Stress including bending and shear (Equation 3-14)*

$$S_e' = 0.5 \cdot S_{ut}$$ *[psi] Endurance Limit (Equation 6-8)*

$$K_a = a \cdot S_{ut}^{\ b}$$ *Surface Factor (Equation 6-19)*

$$K_b = \left[ \frac{D_{Shaft}}{0.3} \right]^{-0.107}$$ *Size Factor for diameters under 2in (Equation 6-20)*

$$K_c = 1$$ *Loading Factor for bending (Equation 6-26)*

$$K_d = 0.975 + 0.000432 \cdot T_s - 0.00000115 \cdot T_s^{\ 2} + 0 \cdot T_s^{\ 3} - 5.95 \times 10^{-13} \cdot T_s^{\ 4}$$ *Temperature Factor at 150C from heated oil (Equation 6-27)*

$$K_e = 1 - 0.08 \cdot 1.645$$ *Reliability Factor of 95% (Equation 6-29)*

$$S_e = K_a \cdot K_b \cdot K_c \cdot K_d \cdot K_e \cdot S_e'$$ *[psi] Marin Equation for endurance limit (Equation 6-18)*

$$n_f = \frac{S_e}{\sigma_{Total}}$$ *Factor of Safety*

SOLUTION

**Unit Settings: Eng F psia mass deg**

$a$ = 1.34 [-]
$A_{Shaft}$ = 0.04909 [in²]
$b$ = -0.085 [-]
$\delta_{Shaft}$ = 0.01 [in]
$\delta_{Spring}$ = 0.15 [in]
$D_{Shaft}$ = 0.25 [in]
E = 2.799E+07 [psi]
Force = 4 [lbf]
I = 0.0001917 [in⁴]
k = 16.77 [lbf/in]
$K_a$ = 0.9287 [-]
$K_b$ = 1.02 [-]
$K_c$ = 1 [-]
$K_d$ = 1.014 [-]
$K_e$ = 0.8684 [-]
L = 4 [in]
$M_z$ = 16 [lbf*in]
$n_f$ = 2.984 [-]
ShaftDeflection = 0.0159 [in]

$\sigma_m$ = 10430 [psi]
$\boxed{\sigma_{Total} = 10.43 \text{ [ksi]}}$
$S_e$ = 31.13 [ksi]
$\overline{S}_{e'}$ = 37.35 [ksi]
$S_{ut}$ = 74.69 [ksi]
$S_y$ = 29.73 [ksi]
$\tau$ = 108.6 [psi]
$T_s$ = 150 [F]

3 potential unit problems were detected.

*Driven Shaft for 5th Axis Rotary*

*This code analyzes the deflection and failure modes for the driven shaft of the 5th axis rotary. This shaft is directly coupled to the gear and is designed to avoid deflecting and shearing.*

*Parameters*

*Geometric Properties*

$D = \frac{5 \text{ [in]}}{16}$ *Shaft diameter*

$N = 30$ *Number of Gear Teeth*

$PitchD = 1.875$ [in] *Pitch Diameter of the Gear*

$L = 0.5$ [in] *Length of unsupported shaft*

*Material Properties*

$E = 3 \times 10^7$ *[psi] Modulus of Elasticity for 1045 steel (Appendix Table A-23)*

$S_y = 32000$ *[psi] Yield Strength (Appendix Table A-22)*

$S_{ut} = 230000$ *[psi] Ultimate Strength (Appendix Table A-22)*

$a = 1.34$ *Ground Surface Finish (Table 6-2)*

$b = -0.085$ *Ground Surface Finish (Table 6-2)*

*Assumptions*

$\nu = 0.5$ [-] *Drive Train Efficiency*

$T_{platter} = 6$ [in*lbf] *Max Torque on the Platter*

$T_f = 150$ [F] *Case Temperature as calculated in Gear Wear*

*Torsional Stress*

$J = \pi \cdot \frac{D^4}{32}$ *[in$^4$] Polar Second Moment of Area (Equation 3-38)*

$\tau_{max} = \frac{T_{platter}}{\nu} \cdot \frac{D}{2 \cdot J}$ *[psi] Maximum Torsional Shear Stress (Equation 3-37)*

*Bending Stress*

$Force = \frac{T_{platter}}{PitchD}$ *[lbf] Force directed on the shaft by the worm*

$M = Force \cdot L$ *[in*lbf] Moment on the shaft*

$\sigma_m = \dfrac{32}{\pi \cdot D^3} \cdot M$ *[psi] Maximum Bending Stress for a solid circular cross section (Equation 3-38)*

*Total Stress*

$\sigma_{Total} = \dfrac{\sigma_m}{2} + \sqrt{\left[\dfrac{\sigma_m}{2}\right]^2 + \tau_{max}^2}$ *[psi] Total Stress including bending and torsion (Equation 3-14)*

$S_e' = 0.5 \cdot S_{ut}$ *[psi] :Endurance Limit (Equation 6-8)*

$K_a = a \cdot S_{ut}^{\,b}$ *Surface Factor (Equation 6-19)*

$K_b = \left[\dfrac{D}{0.3}\right]^{-0.107}$ *Size Factor for diameters under 2in (Equation 6-20)*

$K_c = 1$ *Loading Factor for bending (Equation 6-26)*

$K_d = 0.975 + 0.000432 \cdot T_f - 0.00000115 \cdot T_f^2 + 0 \cdot T_f^3 - 5.95 \times 10^{-13} \cdot T_f^4$ *Temperature Factor at 150C from heated oil (Equation 6-27)*

$K_e = 1 - 0.08 \cdot 1.645$ *Reliability Factor of 95% (Equation 6-29)*

$S_e = K_a \cdot K_b \cdot K_c \cdot K_d \cdot K_e \cdot S_e'$ *[psi] Marin Equation for endurance limit (Equation 6-18)*

$n_f = \dfrac{S_e}{\sigma_{Total}}$ *Factor of Safety*

*Deflection*

$I = \pi \cdot \dfrac{\left[\dfrac{D}{2}\right]^4}{4}$ *[in$^4$] Moment of Inertia for the shaft*

$ShaftDeflection = Force \cdot \dfrac{L^3}{3 \cdot E \cdot I}$ *[in] Resulting Shaft Deflection*

SOLUTION

**Unit Settings: SI C kPa kJ mass deg**

a = 1.34 [-]
b = -0.085 [-]
D = 0.3125 [in]
E = 3.000E+07 [psi]
Force = 3.2 [lbf]
I = 0.0004681 [in$^4$]
J = 0.0009363 [in$^4$]
$K_a$ = 0.4692 [-]
$K_b$ = 0.9956 [-]
$K_c$ = 1 [-]
$K_d$ = 1.014 [-]
$K_e$ = 0.8684 [-]

$L$ = 0.5 [in]
$M$ = 1.6 [in*lbf]
$N$ = 30 [-]
$\nu$ = 0.5 [-]
$n_f$ = 20.67 [-]
PitchD = 1.875 [in]
ShaftDeflection = 0.000009494 [in]
$\sigma_m$ = 534 [psi]
$\sigma_{Total}$ = 2287 [psi]
$S_e$ = 47288 [psi]
$S_{e'}$ = 115000 [psi]
$S_{ut}$ = 230000 [psi]
$S_y$ = 32000 [psi]
$\tau_{max}$ = 2003 [psi]
$T_f$ = 150 [F]
$T_{platter}$ = 6 [in*lbf]

3 potential unit problems were detected.

*Moment of Inertia for 4th and 5th Axis Rotary Table*

*This code solves for load moment of inertia that the motors will experience while driving the rotary. Each axis is input differently due to different loads. Reflected moment of inertia is included with shaft and coupling inertia to find the system inertia. Axis A inertia is calculated using simplified geometry for rotary stage B.*

*Axis B Parameters*

$H_{part}$ = 0.1 [m] *Part Height*

$D_{part}$ = 0.1 [m] *Part Diameter*

$\rho_{part}$ = 900 [kg/m$^3$] *Part Density*

$H_{platter}$ = 0.07 [m] *Table Height*

$D_{platter}$ = 0.1 [m] *Platter Diameter*

$\rho_{platter}$ = 8000 [kg/m$^3$] *Table Density*

$D_{shaft}$ = 0.0047625 [m] *Shaft Diameter (3/16in)*

$L_{shaft}$ = 0.1 [m] *Shaft Length*

$\rho_{shaft}$ = 8000 [kg/m$^3$] *Shaft Density*

$D_{coupling}$ = 0.015 [m] *Coupling Diameter*

$L_{coupling}$ = 0.0508 [m] *Coupling Length*

$\rho_{coupling}$ = 2700 [kg/m$^3$] *Coupling Density*

$D_{gear}$ = 0.0254 [m] *30 Tooth Gear Diameter*

$H_{gear}$ = 0.00762 [m] *Gear Width*

$\rho_{gear}$ = 8500 [kg/m$^3$] *Brass Gear Density*

$N_B$ = 30 [-] *Gear Teeth*

*Axis A Parameters*

$m_B$ = 7 [kg] *Axis B Mass*

$H_B$ = 0.0762 [m] *Axis B Height*

$L_B$ = 0.1524 [m] *Axis B Length*

$W_B$ = 0.1524 [m] *Axis B Width*

$Offset_B$ = 0.1 [m] *Axis B offset to Axis A*

$M_{Motor,B}$ = 0.5 [kg] *Axis B Motor Mass*

$W_{Motor,B} = 0.04$ [m] *Axis B Motor Width*

$N_A = 30$ *Gear Teeth*

$B_{SolidWorks} = 10$ *SolidWorks calculated inertia of axis B assembly*

*Solve for Axis B Inertia*

$$M_{part} = \pi \cdot \left[ \frac{D_{part}}{2} \right]^2 \cdot H_{part} \cdot \rho_{part}$$ *[kg] Part Mass*

$$I_{part} = \frac{M_{part}}{8} \cdot D_{part}^{\ 2}$$ *[kg*m²] Part Inertia*

$$M_{platter} = \pi \cdot \left[ \frac{D_{platter}}{2} \right]^2 \cdot H_{platter} \cdot \rho_{platter}$$ *[kg] Platter Mass*

$$I_{platter} = \frac{M_{platter}}{8} \cdot D_{platter}^{\ 2}$$ *[kg*m² Platter Inertia*

$$M_{gear} = \pi \cdot \left[ \frac{D_{gear}}{2} \right]^2 \cdot H_{gear} \cdot \rho_{gear}$$ *[kg] Gear Mass*

$$I_{gear} = \frac{M_{gear}}{8} \cdot D_{gear}^{\ 2}$$ *[kg*m²] Gear Inertia*

$$I_{Reflected,B} = \frac{I_{part} + I_{platter} + I_{gear}}{N_B^{\ 2}}$$ *Find the Reflected Inertia*

$$M_{shaft} = \pi \cdot \left[ \frac{D_{shaft}}{2} \right]^2 \cdot L_{shaft} \cdot \rho_{shaft}$$ *[kg] Shaft Mass*

$$I_{shaft} = \frac{M_{shaft}}{8} \cdot D_{shaft}^{\ 2}$$ *[kg*m²] Shaft Inertia*

$$M_{coupling} = \pi \cdot \left[ \frac{D_{coupling}}{2} \right]^2 \cdot L_{coupling} \cdot \rho_{coupling}$$ *[kg] Coupling Mass*

$$I_{coupling} = \frac{M_{coupling}}{8} \cdot D_{coupling}^{\ 2}$$ *[kg*m²] Coupling Inertia*

$$Inertia_{Total,B} = I_{Reflected,B} + I_{shaft} + I_{coupling}$$ *Total Inertia Experienced by Motor at B-axis*

*Solve for Axis A Inertia*

$$I_B = \frac{m_B}{12} \cdot \left[ L_B^{\ 2} + H_B^{\ 2} \right] + m_B \cdot Offset_B^{\ 2}$$ *[kg*²] Inertia for Stage B Axis Through A Axis*

$$I_{Part,A} = \frac{M_{part}}{12} \cdot \left[ 3 \cdot \left( \frac{D_{part}}{2} \right)^2 + H_{part}^{\ 2} \right] + M_{part} \cdot \left[ H_{part} - Offset_B \right]^2$$ *[kg*m²] Inertia for the Part*

$$I_{motor,B} = \frac{M_{Motor,B}}{12} \cdot \left[ 2 \cdot W_{Motor,B} \right]^2 + M_{Motor,B} \cdot Offset_B^{\ 2}$$ *[kg*m$^2$] Inertia for Stage B Motor Through A Axis*

$$I_{Reflected,A} = \frac{I_B + I_{Part,A} + I_{motor,B} + I_{gear}}{N_A^{\ 2}}$$ *[kg*m$^2$] Reflected Inertia Experienced by Motor A*

$$Inertia_{total,A} = I_{Reflected,A} + I_{shaft} + I_{coupling}$$ *[kg*m$^2$]*

SOLUTION

**Unit Settings: SI C kPa kJ mass deg**

$B_{SolidWorks}$ = 10 [kg*m$^2$]
$D_{coupling}$ = 0.015 [m]
$D_{gear}$ = 0.0254 [m]
$D_{part}$ = 0.1 [m]
$D_{platter}$ = 0.1 [m]
$D_{shaft}$ = 0.004763 [m]
$H_B$ = 0.0762 [m]
$H_{gear}$ = 0.00762 [m]
$H_{part}$ = 0.1 [m]
$H_{platter}$ = 0.07 [m]
$Inertia_{total,A}$ = 0.0001043 [kg*m$^2$]
$Inertia_{Total,B}$ = 0.000007815 [kg*m$^2$]
$I_B$ = 0.08694 [kg*m$^2$]
$I_{coupling}$ = 6.817E-07 [kg*m$^2$]
$I_{gear}$ = 0.000002647 [kg*m$^2$]
$I_{motor,B}$ = 0.005267 [kg*m$^2$]
$I_{part}$ = 0.0008836 [kg*m$^2$]
$I_{Part,A}$ = 0.001031 [kg*m$^2$]
$I_{platter}$ = 0.005498 [kg*m$^2$]
$I_{Reflected,A}$ = 0.0001036 [kg*m$^2$]
$I_{Reflected,B}$ = 0.000007093 [kg*m$^2$]
$I_{shaft}$ = 4.040E-08 [kg*m$^2$]
$L_B$ = 0.1524 [m]
$L_{coupling}$ = 0.0508 [m]
$L_{shaft}$ = 0.1 [m]
$m_B$ = 7 [kg]
$M_{coupling}$ = 0.02424 [kg]
$M_{gear}$ = 0.03282 [kg]
$M_{Motor,B}$ = 0.5 [kg]
$M_{part}$ = 0.7069 [kg]
$M_{platter}$ = 4.398 [kg]
$M_{shaft}$ = 0.01425 [kg]
$N_A$ = 30 [-]
$N_B$ = 30 [-]
$Offset_B$ = 0.1 [m]
$\rho_{coupling}$ = 2700 [kg/m$^3$]
$\rho_{gear}$ = 8500 [kg/m$^3$]
$\rho_{part}$ = 900 [kg/m$^3$]
$\rho_{platter}$ = 8000 [kg/m$^3$]
$\rho_{shaft}$ = 8000 [kg/m$^3$]
$W_B$ = 0.1524 [m]
$W_{Motor,B}$ = 0.04 [m]

```
"Worm Gear Analysis"

"This code verifies the rated power for worm gears. From input parameters, this program predicts life time of a worm gear.
Additionally, the gear case is analyzed for oil temperature. Values are found from eqautions in the Shigley engineering book"

"Conversions"
Conv_hp = 746 [W/hp]

 "Geometry Parameters"
PitchDiameter_G = 1.875 [in] "Gear Pitch Diameter"
PitchDiameter_W = 0.625 [in] "Worm Pitch Diameter"
Speed = 100 [rev/min] "Speed of the Gear"
N_G = 30 [-] "Number of gear teeth"
N_W = 1 [-] "Number of worm teeth"
P_in = 200[W] "Input Power"
P_out = 90[W] "Required Output Power"
dp = 16 [1/in] "Diametral Pitch of the gear"
K_w = 100 [-] "Worm gear wear factor (Table 15-11)"
t_Face = 0.312 [in] "Effective Face Width"
PA = 14.5 [deg] "Normal Pressure Angle"
K_a = 1.5 [-] "Application Factor (Table 15-2)"
A_Case = 15 [in^2] "Gear Case Lateral Area"
T_a = 70 [F] "Atmospheric Temperature"

C_s = 1000 [-] "Materials Factor for Chilled Cast Gears (Equation 15-34)"
K_w_Buck = 60 [-] "Worm Gear Wear Factor From Buckingham (Table 15-11)"

"AGMA Method"
p_x = pi/dp "[in] Axial Pitch"
L = p_x*N_W "Lead of the worm"
gamma = ARCTAN(L/(pi*PitchDiameter_W)) "[deg] Lead Angle of the worm"

p_n = pi/(dp/cos(gamma)) "[in] Normal Diametral Pitch"

V_s = pi*PitchDiameter_W*Speed*N_G "[in/min] Sliding Velocity"
f = 0.103*exp(-0.110*(V_s^0.450))+0.012 "Friction Factor (Equation 15-62)"
e = (cos(PA)-f*tan(gamma))/(cos(PA) + f/tan(gamma)) "Calculate the Efficiency (Equation 15-58)"

V_G = Speed*pi*PitchDiameter_G "[in/min] Gear Tangential Speed"

W_g_t = 33000*P_out*0.001341*K_a/(V_G*e) "[lbf] Tooth load (Equation 15-58)"

C_m = 0.0107*sqrt(-N_G^2+56*N_G+5145) "Ratio Correction Factor (Equation 15-37)"
C_v = 62.52*V_s^(-0.774) "Velocity Correction Factor (Equation 15-38)"

W_t_all = C_s*(N_G/dp)^0.8*t_Face*C_m*C_v "[lbf] Allowable Tangential Force on Worm (Equation 15-28)"

W_f = f*W_g_t/(f*sin(gamma)-cos(14.5[deg])*cos(gamma)) "[lbf] Friction Force (Equation 15-61)"
H_f = abs(W_f)*v_s*0.001341/33000 "[W] Power Dissapated in Frictional Work (Equation 15-63)"

Sigma_G = W_g_t/(p_n*t_Face*0.1) "[psi] Bending Stress in a Gear Tooth from Buckingham adapted Lewis Equation and
form Facor {Equation 15-53)"

n_f = W_t_all/W_g_t "Factor of Safety for 25000 hours"

"Buckingham Wear Load"
W_Buck = K_w_Buck*PitchDiameter_G*t_Face "[lbf] Allowable gear loading for wear (Equation 15-64)"

"Gear Case Temperature"
H_loss = 33000*(1-e)*P_in*0.001341 "[ft*lbf/min] Heat Loss Rate from the Worm Gear Case (Equation 15-49)"
h_CR = Speed*N_G/6494+0.13 " [ft*lbf/(min*in^2*F)] Heat Transfer Coefficient for Convection and Radiation (Equation 15-50)
"
```

t_s = T_a + H_loss/(h_CR*A_Case) "[F] Oil Sump Temperature (Equation 15-51)"

A_min = 43.2*((PitchDiameter_W+PitchDiameter_G)/2)^1.7 "[in^2] AGMA Recommended Minimum Lateral Area (Equation 15-52)"

SOLUTION

**Unit Settings: Eng F psia mass deg**

$A_{Case}$ = 15 [$in^2$]
$A_{min}$ = 63.13 [$in^2$]
$Conv_{hp}$ = 746 [W/hp]
$C_m$ = 0.8236 [-]
$C_s$ = 1000 [-]
$C_v$ = 0.0755 [-]
dp = 16 [1/in]
e = 0.885 [-]
f = 0.01243 [-]
$\gamma$ = 5.711 [deg]
$h_{CR}$ = 0.592 [ft*lbf/(min*$in^2$*F)]
$H_f$ = 0.00003545
$H_{loss}$ = 1017 [ft*lbf/min]
$K_a$ = 1.5 [-]
$K_w$ = 100 [-]
$K_{w,Buck}$ = 60 [-]
L = 0.1963 [in]
$n_f$ = 2.799 [-]
$N_G$ = 30 [-]
$N_W$ = 1 [-]
PA = 14.5 [deg]
$PitchDiameter_G$ = 1.875 [in]
$PitchDiameter_W$ = 0.625 [in]
$P_{in}$ = 200 [W]
$p_n$ = 0.1954 [in]
$P_{out}$ = 90 [W]
$p_x$ = 0.1963 [in]
$\sigma_G$ = 1880 [psi]
Speed = 100 [rev/min]
$T_a$ = 70 [F]
$t_{Face}$ = 0.312 [in]
$t_s$ = 184.6 [F]
$V_G$ = 589 [in/min]
$V_s$ = 5890 [in/min]
$W_{Buck}$ = 35.1 [lbf]
$W_f$ = -0.1481 [lbf]
$W_{g,t}$ = 11.46 [lbf]
$W_{t,all}$ = 32.08 [lbf]

11 potential unit problems were detected.

KEY VARIABLES

$n_f$ = 2.799 [-] *Factor of Safety for 25000 hours*

```
"Worm Gear Analysis"

"This code verifies the rated power for worm gears. From input parameters, this program predicts life time of a worm gear.
Additionally, the gear case is analyzed for oil temperature. Values are found from eqautions in the Shigley engineering book"

"Conversions"
Conv_hp = 746 [W/hp]

 "Geometry Parameters"
PitchDiameter_G = 1.875 [in] "Gear Pitch Diameter"
PitchDiameter_W = 0.625 [in] "Worm Pitch Diameter"
Speed = 100 [rev/min] "Speed of the Gear"
N_G = 30 [-] "Number of gear teeth"
N_W = 1 [-] "Number of worm teeth"
P_in = 100[W] "Input Power"
P_out = 50[W] "Required Output Power"
dp = 16 [1/in] "Diametral Pitch of the gear"
K_w = 100 [-] "Worm gear wear factor (Table 15-11)"
t_Face = 0.312 [in] "Effective Face Width"
PA = 14.5 [deg] "Normal Pressure Angle"
K_a = 1.5 [-] "Application Factor (Table 15-2)"
A_Case = 15 [in^2] "Gear Case Lateral Area"
T_a = 70 [F] "Atmospheric Temperature"

C_s = 1000 [-] "Materials Factor for Chilled Cast Gears (Equation 15-34)"
K_w_Buck = 60 [-] "Worm Gear Wear Factor From Buckingham (Table 15-11)"

"AGMA Method"
p_x = pi/dp "[in] Axial Pitch"
L = p_x*N_W "Lead of the worm"
gamma = ARCTAN(L/(pi*PitchDiameter_W)) "[deg] Lead Angle of the worm"

p_n = pi/(dp/cos(gamma)) "[in] Normal Diametral Pitch"

V_s = pi*PitchDiameter_W*Speed*N_G "[in/min] Sliding Velocity"
f = 0.103*exp(-0.110*(V_s^0.450))+0.012 "Friction Factor (Equation 15-62)"
e = (cos(PA)-f*tan(gamma))/(cos(PA) + f/tan(gamma)) "Calculate the Efficiency (Equation 15-58)"

V_G = Speed*pi*PitchDiameter_G "[in/min] Gear Tangential Speed"

W_g_t = 33000*P_out*0.001341*K_a/(V_G*e) "[lbf] Tooth load (Equation 15-58)"

C_m = 0.0107*sqrt(-N_G^2+56*N_G+5145) "Ratio Correction Factor (Equation 15-37)"
C_v = 62.52*V_s^(-0.774) "Velocity Correction Factor (Equation 15-38)"

W_t_all = C_s*(N_G/dp)^0.8*t_Face*C_m*C_v "[lbf] Allowable Tangential Force on Worm (Equation 15-28)"

W_f = f*W_g_t/(f*sin(gamma)-cos(14.5[deg])*cos(gamma)) "[lbf] Friction Force (Equation 15-61)"
H_f = abs(W_f)*v_s*0.001341/33000 "[W] Power Dissapated in Frictional Work (Equation 15-63)"

Sigma_G = W_g_t/(p_n*t_Face*0.1) "[psi] Bending Stress in a Gear Tooth from Buckingham adapted Lewis Equation and
form Facor {Equation 15-53)"

n_f = W_t_all/W_g_t "Factor of Safety for 25000 hours"

"Buckingham Wear Load"
W_Buck = K_w_Buck*PitchDiameter_G*t_Face "[lbf] Allowable gear loading for wear (Equation 15-64)"

"Gear Case Temperature"
H_loss = 33000*(1-e)*P_in*0.001341 "[ft*lbf/min] Heat Loss Rate from the Worm Gear Case (Equation 15-49)"
h_CR = Speed*N_G/6494+0.13 " [ft*lbf/(min*in^2*F)] Heat Transfer Coefficient for Convection and Radiation (Equation 15-50)
"
```

t_s = T_a + H_loss/(h_CR*A_Case) "[F] Oil Sump Temperature (Equation 15-51)"

A_min = 43.2*((PitchDiameter_W+PitchDiameter_G)/2)^1.7 "[in^2] AGMA Recommended Minimum Lateral Area (Equation 15-52)"

SOLUTION

**Unit Settings: Eng F psia mass deg**

$A_{Case}$ = 15 [in$^2$]
$A_{min}$ = 63.13 [in$^2$]
$Conv_{hp}$ = 746 [W/hp]
$C_m$ = 0.8236 [-]
$C_s$ = 1000 [-]
$C_v$ = 0.0755 [-]
dp = 16 [1/in]
e = 0.885 [-]
f = 0.01243 [-]
$\gamma$ = 5.711 [deg]
$h_{CR}$ = 0.592 [ft*lbf/(min*in$^2$*F)]
$H_f$ = 0.00001969
$H_{loss}$ = 508.7 [ft*lbf/min]
$K_a$ = 1.5 [-]
$K_w$ = 100 [-]
$K_{w,Buck}$ = 60 [-]
L = 0.1963 [in]
$n_f$ = 5.039 [-]
$N_G$ = 30 [-]
$N_W$ = 1 [-]
PA = 14.5 [deg]
$PitchDiameter_G$ = 1.875 [in]
$PitchDiameter_W$ = 0.625 [in]
$P_{in}$ = 100 [W]
$p_n$ = 0.1954 [in]
$P_{out}$ = 50 [W]
$p_x$ = 0.1963 [in]
$\sigma_G$ = 1044 [psi]
Speed = 100 [rev/min]
$T_a$ = 70 [F]
$t_{Face}$ = 0.312 [in]
$t_s$ = 127.3 [F]
$V_G$ = 589 [in/min]
$V_s$ = 5890 [in/min]
$W_{Buck}$ = 35.1 [lbf]
$W_f$ = -0.08228 [lbf]
$W_{g,t}$ = 6.366 [lbf]
$W_{t,all}$ = 32.08 [lbf]

11 potential unit problems were detected.

KEY VARIABLES

$n_f$ = 5.039 [-] *Factor of Safety for 25000 hours*

```
"Bearings"
"This code solves for bearing performance."

"NSK 7006CTYNDBL P4 ABEC-7"
"Basic Load Ratings"
C_r = 3400 [lbf] "Basic Dynamic Load Rating"
C_or = 2310 [lbf] "Basic Static Load Rating"
C_axial = 1540 [lbf] "Permissible Axial Load"

"Geometrical Parameters"
a = 0.48 [in] "Effective Load Center"

"Design Parameters"
h_part = 4 [in] "Part Height"
F_design = 18 [lbf] "80 N Cutting Force"
m_B = 20 [lbf] "Weight of axis"
L_B = 6 [in] "Length of B Axis"
N_design = 100 [rev/min] "Platter Rotational Speed"

"Limiting Speeds"
N|grease_max = 27100 [rev/min] "Maximum rated speed for greased bearings"
N|oil_max = 41200 [rev/min] "Maximum rated speed for oiled bearings"

"Rated Load"
M_rated = a * C_or "[lbf-in] Rated Moment"

"Design Load"
M|B_design = h_part * F_design "[lbf-in] Axis B Design Moment"
M|A_design = F_design * L_B + m_B * L_B/2 "[lbf-in] Axis A Design Moment"

"Factors of Safety"
eta|B_M = M_rated/M|B_design "Axis B Factor of Safety for Bending Moment"
eta|B_A = C_axial/F_design "Axis B Factor of Safety for Axial Loading"

eta|A_M = M_rated/M|A_design "Axis A Factor of Safety for Bending Moment"
eta|A_A = C_axial / (m_B + F_design) "Axis A Factor of Safety for Axial Loading"

eta_N = N|grease_max/N_design "Factor of Safety for Speed Rating"
```

SOLUTION

**Unit Settings: Eng F psia mass deg**

$a$ = 0.48 [in]
$C_{axial}$ = 1540 [lbf]
$C_{or}$ = 2310 [lbf]
$C_r$ = 3400 [lbf]
$\eta_N$ = 271 [-]
$\eta^A_A$ = 40.53 [-]
$\eta^A_M$ = 6.6 [-]
$\eta^B_A$ = 85.56 [-]
$\eta^B_M$ = 15.4 [-]
$F_{design}$ = 18 [lbf]
$h_{part}$ = 4 [in]
$L_B$ = 6 [in]
$m_B$ = 20 [lbf]
$M_{rated}$ = 1109 [lbf*in]
$M^A_{design}$ = 168 [lbf*in]
$M^B_{design}$ = 72 [lbf*in]
$N_{design}$ = 100 [rev/min]

$N_{max}^{grease}$ = 27100 [rev/min]
$N_{max}^{oil}$ = 41200 [rev/min]

No unit problems were detected.

*Using 1/4-20 bolts, Grade 1*

$F = 45$ [lbf] *weight+cutting force*

$M = 45 \cdot 3$ [in-lbf] *Moment*

$t = 0.5$ [in] *fixture plate thickness*

$d = 0.25$ [in] *bolt nominal diameter*

$Area = \pi \cdot \frac{d^2}{4}$ *bolt area*

$n = 4$ *number of bolts*

$S_{y,mem} = 40000$ [psi] *6061 Al - Yield Strength*

$S_{ut,mem} = 45000$ [psi] *6061 Al - Yield Strength*

$S_y = 36000$ [psi] *Grade 1 Steel - Yield Strength (Table 8-9)*

$S_p = 33000$ [psi] *Grade 1 Steel - Proof Strength (Table 8-9)*

$S_{ut} = 60000$ [psi] *Grade 1 Steel - Ultimate Tensile Strength (Table 8-9)*

*Fixture plate attaching to Axis B*

*Bending Stress*

$\sigma_{bend} = M \cdot \frac{c}{I}$

$I = \frac{1}{12} \cdot b \cdot h^3$ *Moment of Inertia*

$b = 6$ [in] *Base of Housing*

$h = 3$ [in] *Height of Housing*

$c = 1.5$ [in] *distance from neutral axis*

$a = 1$ [in] *distance from edge*

$\eta_{B,bend} = \frac{S_y}{\sigma_{bend}}$ *Bending Stress Safety Factor for B*

*Bearing stress*

$\sigma_b = \frac{F}{t \cdot d}$

$\eta_{B,bearing} = \frac{S_y}{\sigma_b}$ *Bearing Stress Safety Factor for B*

*Shear stress*

$$\tau = \frac{F}{4 \cdot \pi \cdot \frac{d^2}{4}}$$

$$\eta_{B,shear} = 0.577 \cdot \frac{S_y}{\tau}$$ *Shear Stress Safety Factor for B*

*Edge shearing of member*

$$\tau_{edge} = \frac{F}{4 \cdot a \cdot t}$$

$$\eta_{B,edge} = 0.577 \cdot \frac{S_{y,mem}}{\tau_{edge}}$$ *Edge shearing of member Safety Factor for B*

*Bolt pattern attaching fixture plate to Axis A*

$r = 3$ [in] *radii of bolt pattern*

$$F_1 = \frac{F}{n}$$ *primary shear load*

$$F_2 = M \cdot \frac{r}{4 \cdot r^2}$$ *secondary shear load*

$\alpha = 30$ [deg] *90 to 180 deg. first bolt*

$F_{A,x} = F_2 \cdot \mathbf{sin}[\alpha]$ *x component of bolt*

$F_{A,y} = F_1 + F_2 \cdot \mathbf{cos}[\alpha]$ *y component of bolt*

$F_A = \sqrt{F_{A,x}^2 + F_{A,y}^2}$ *Total force on bolt*

$\beta = 60$ [deg] *180 deg to 270 deg*

$F_{C,x} = F_2 \cdot \mathbf{sin}[\beta]$ *x component of bolt*

$F_{C,y} = F_2 \cdot \mathbf{cos}[\beta] - F_1$ *y component of bolt*

$F_C = \sqrt{F_{C,x}^2 + F_{C,y}^2}$ *Total force on bolt*

*largest stress at 0 deg*

$F_B = F_1 + F_2$ *The largest stress is seen when rotated 90 deg with bolt at 0 deg*

*shear stress*

$$\tau_A = \frac{F_B}{Area}$$

$$\eta_{A,shear} = 0.577 \cdot \frac{S_y}{\tau_A}$$

Bearing stress

$$\sigma_{A,b} = \frac{-F_B}{Area}$$

$$\eta_{A,b} = \frac{S_y}{\sigma_{A,b}}$$ *Bearing Stress Safety Factor for A*

*Critical bending stress*

$$I_b = \bar{I} + 3 \cdot \left[ I_{holes} + r^2 \cdot 0.25 \text{ [in]} \cdot 0.25 \text{ [in]} \right]$$ *Total moment of inertia, Area is bolt diam*thickness*

$$\bar{I} = 0.5 \cdot \frac{4^3}{12} \cdot 1 \text{ [in}^4\text{]}$$ *Cross section inertia of fixture plate*

$$I_{holes} = 0.5 \cdot \frac{0.25^3}{12} \cdot 1 \text{ [in}^4\text{]}$$ *Inertia of holes*

$$\sigma_{A,bend} = M \cdot \frac{2 \text{ [in]}}{I_b}$$

$$\eta_{A,bend} = \frac{S_y}{\sigma_{A,bend}}$$ *Critical Bending Stress Safety Factor for A*

$A_t = 0.031$ *Tensile stress area*

$A_s = 0.25 \cdot E_s$ *Shear area of screw*

$A_n = 0.25 \cdot K_n$ *Shear area of hole*

$K_n = 0.1876$ [psi] *Internal minor diameter of hole*

$n_t = 20$ *Number of threads*

$E_s = 0.2164$ *Min pitch diameter of external threads*

$$L_e = 2 \cdot \frac{A_t}{K_n \cdot \pi \cdot \left[ 0.5 + 0.577 \cdot n \cdot (E_s - K_n) \right]}$$ *Length of engagement*

$$J = A_s \cdot \frac{S_{ut}}{A_n \cdot S_{ut,mem}}$$ *For screwing steel into aluminum*

$L_{ef} = L_e \cdot J$ *New length of engagement*

SOLUTION

**Unit Settings: Eng F psia mass deg**

$a = 1$ [in]
Area = 0.04909 [in$^2$]
$A_s = 0.0541$ [psi]
$b = 6$ [in]
$c = 1.5$ [in]
$\alpha = 30$ [deg]
$A_n = 0.0469$ [psi]
$A_t = 0.031$ [psi]
$\beta = 60$ [deg]

$\eta_{A,b}$ = -78.54
$\eta_{A,bend}$ = 580.8
$\eta_{A,shear}$ = 45.32
$\eta_{B,bearing}$ = 100
$\eta_{B,bend}$ = 2400
$\eta_{B,edge}$ = 1026
$\eta_{B,shear}$ = 90.63
$E_s$ = 0.2164 [in]
F = 45 [lbf]
$F_1$ = 11.25 [lbf]
$F_2$ = 11.25 [lbf]
$F_A$ = 21.73 [lbf]
$F_{A,x}$ = 5.625 [lbf]
$F_{A,y}$ = 20.99 [lbf]
$F_B$ = 22.5 [lbf]
$F_C$ = 11.25 [lbf]
$F_{C,x}$ = 9.743 [lbf]
$F_{C,y}$ = -5.625 [lbf]
h = 3 [in]
I = 13.5 [$in^4$]
$I_b$ = 4.356 [$in^4$]
$\bar{I}$ = 2.667 [$in^4$]
$I_{holes}$ = 0.000651 [$in^4$]
J = 1.538 [-]
$K_n$ = 0.1876 [psi]
$L_e$ = 0.1857 [in]
$L_{ef}$ = 0.2856 [in]
M = 135 [lbf*in]
n = 4
$n_t$ = 20
r = 3 [in]
$\sigma_{A,b}$ = -458.4 [psi]
$\sigma_{A,bend}$ = 61.98 [psi]
$\sigma_b$ = 360 [psi]
$\sigma_{bend}$ = 15 [psi]
$S_p$ = 33000 [psi]
$S_{ut}$ = 60000 [psi]
$S_{ut,mem}$ = 45000 [psi]
$S_y$ = 36000 [psi]
$S_{y,mem}$ = 40000 [psi]
t = 0.5 [in]
$\tau$ = 229.2 [psi]
$\tau_A$ = 458.4 [psi]
$\tau_{edge}$ = 22.5 [psi]

2 potential unit problems were detected.

# Appendix F

DVP&R

FMEA

# ME428/ME481 DVP&R Format

Report Date: 1/27/2016 | Sponsor: IME Dept (Jose Macedo / Martin Koch) | Component/Assembly: Rotary Table | REPORTING ENGINEER:

## TEST PLAN / TEST REPORT

| Item No | Specification or Clause Reference | Test Description | Acceptance Criteria | Test Responsibility | Test Stage | SAMPLES | | TIMING | | TEST RESULTS | | | NOTES |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Quantity | Type | Start date | Finish date | Test Result | Quantity Pass | Quantity Fail | |
| 1 | Specification 3.5 | Vibrations Test | Natural frequency > 2500 Hz | Ricky | DV | 1 | B | 2/8/2016 | 2/18/2016 | Fail | 0 | 1 | Frequency ~400Hz, conclusive |
| 2 | Specification 12.2 | Gear Wear Test | life cycle > 2400 Hrs | Dakota | DV | 1 | B | - | - | - | - | - | Incomplete |
| 3 | Specification 2.4 | Backlash Test | Gears must perfectly mesh | Irene | CV | 1 | A | 5/12/2016 | 5/15/2016 | Pass | 1 | 0 | Manually adjust spring plunger to mesh |
| 4 | Specification 2.3 | Accuracy Test (Axis B) | Resolution ≤ 0.001 deg | Dakota | DV | 1 | B | - | - | - | - | - | Incomplete |
| 5 | Specification 2.3 | Accuracy Test (Axis A) | Resolution ≤ 0.001 deg | Irene | DV | 1 | B | - | - | - | - | - | Incomplete |
| 6 | | Oil Temperature Test | Temperature of oil < | Nicole | PV | 1 | C | - | - | - | - | - | Incomplete |
| 7 | | Oil Leakage Test | No leaks | Irene | CV | 1 | A | - | - | - | - | - | Incomplete |
| 8 | | Motor Torque Curves | Curves match given | Nicole | PV | 1 | C | | | | | | Motors changed, could not test |
| 9 | | Power Requirement | Works with Haas Machine | Ricky | PV | 1 | C | | | | | | Incomplete |
| 10 | Specification 9.1 | Tolerances / Runnout | Tolerance < 0.01 in | Ricky | DV | 1 | B | 3/5/2016 | 3/7/2016 | Pass | 1 | 0 | Dimensions checked with Optical Comparator, deviations of 0.002 |
| 11 | Specification 2.1 | Speed Test (Axis B) | Speed > 30 rpm | Nicole | DV | 1 | B | 4/31/2016 | 5/1/2016 | Pass | 1 | 0 | |
| 12 | Specification 2.2 | Speed Test (Axis A) | Speed > 30 rpm | Nicole | DV | 1 | B | 4/31/2016 | 5/1/2016 | Pass | 1 | 0 | |

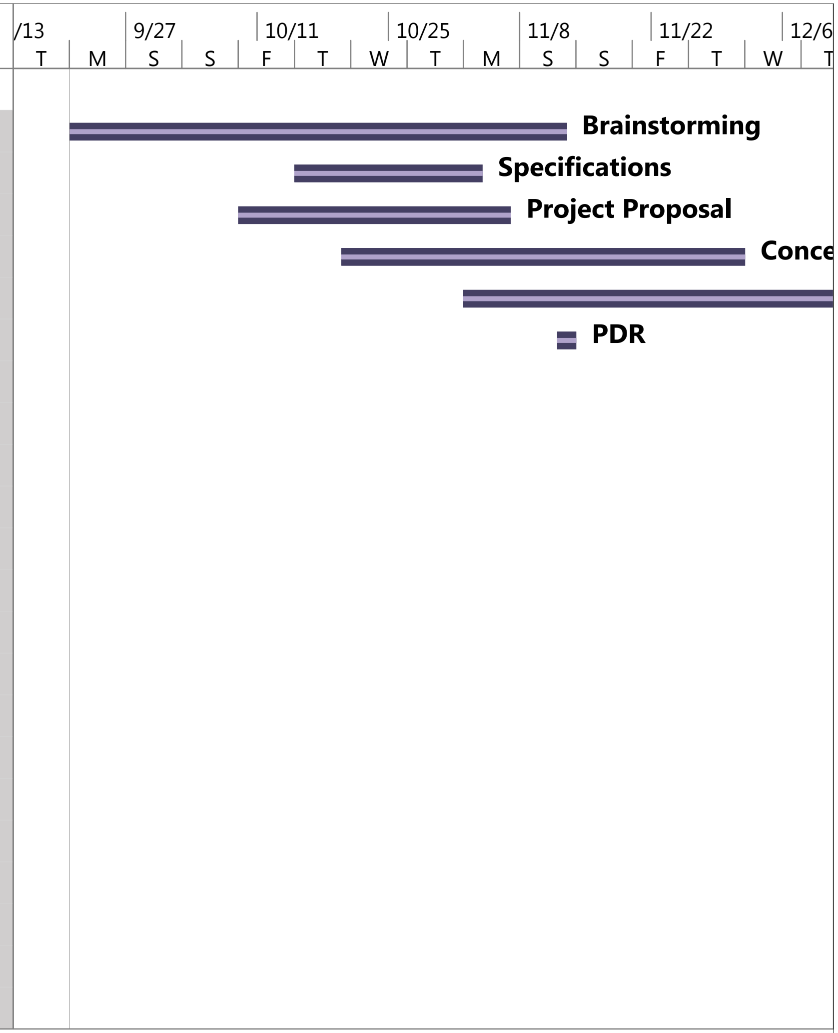
# Failure Mode and Effect Analysis

| Component / Function | Potential Failure Mode | Potential Cause(s) Mechanism(s) of Failure | Local Effects of Failure | Next higher level effect | System level end effect | Probability | Severity | Detection | Risk Priority Number | Actions Taken |
|---|---|---|---|---|---|---|---|---|---|---|
| Housing | Vibration | Natural Frequency | Chatter | Parts could loosen or shift | Loss of tight tolerances | 1 | 5 | 1 | 5 | Calculations and testing performed |
| Platter | Loosen from shaft | Natural Frequency | Platter could shift | Machine part will not be properly centered | Loss of tight tolerances | 1 | 5 | 1 | 5 | Calculations and testing performed |
| Housing Cap | Loosen from housing | Natural Frequency | Cap could shift | Bearings could be preloaded improperly | Ease of movement will decrease | 1 | 5 | 1 | 5 | Calculations and testing performed |
| Platter O-ring | Break or fall out of place | Wear | Particles can go past platter | Bearings can be contaminated | Ease of movement will decrease | 4 | 3 | 1 | 12 | - |
| Angular Ball Bearing | Corrosion | Contamination, Wear | Bearing will be harder to turn | System will have more friction | More wear on gears and motor, possible lack of movement | 3 | 7 | 4 | 84 | Calculations |
| Worm+Gear | Teeth Shear | External Forces, Wear | Gears will not mesh properly | System will have more friction | Possible complete gear failure: rotary unusable | 4 | 7 | 2 | 56 | Calculations and testing performed |
| Worm Shaft | Shear | External Forces | Shaft is deformed or broken | Motor cannot transmit power | Rotary motion impaired, or unusable | 3 | 7 | 1 | 21 | Calculations |
| Gear Shaft | Shear | External Forces | Shaft is deformed or broken | Motor cannot transmit power | Rotary motion impaired, or unusable | 3 | 6 | 1 | 18 | Calculations |
| Yaskawa Motor | Electronic | Incorrect Power | Motor will not function | Axis will be unable to spin | Rotary motion impaired, or unusable | 1 | 7 | 1 | 7 | - |
| Motor Coupler | Loosen from shafts | External Forces, Wear | Shaft will not be connected to motor | Motor cannot transmit power | Rotary speed impaired | 2 | 7 | 1 | 14 | - |
| Worm Shaft Bearing | Corrosion | Contamination, Wear | Bearing will be harder to turn | System will have more friction | Ease of movement will decrease | 7 | 2 | 4 | 56 | Calculations |
| Gear Preload Bearing | Corrosion | Contamination, Wear | Bearing will be harder to turn | System will have more friction | Ease of movement will decrease | 7 | 2 | 4 | 56 | Calculations |
| Spring | Shear | External Forces | Spring force will decrease | Possible improper preload on gears | Backlash will reduce tolerances | 5 | 4 | 2 | 40 | Calculations |
| Spring Enforcer Linear Bearing | Corrosion | Contamination, Wear | Bearing rollers will be harder to spin | System will have more friction | Possible misalignment of gear preload | 2 | 4 | 4 | 32 | Calculations |
| Plastic Oil Cover | Crack or leak | Wear | Oil will leave gear area | System could have more friction | More wear on gears and motor | 4 | 1 | 1 | 4 | - |
| Rotary Connection Pins | Shear | External Forces | Pin(s) will no longer Provide support | Axis A will be detached or less supported | Rotary unusable | 2 | 10 | 1 | 20 | - |

# Appendix G

Gantt Chart

| ID | | Task Mode | Task Name | Duration | Start | Finish |
|---|---|---|---|---|---|---|
| 1 | | | | | | |
| 2 | | | Brainstorming | 39 days | Mon 9/21/15 | Thu 11/12/15 |
| 3 | | | Specifications | 14 days | Thu 10/15/15 | Tue 11/3/15 |
| 4 | | | Project Proposal | 21 days | Fri 10/9/15 | Fri 11/6/15 |
| 5 | | | Concept Generation | 31 days | Tue 10/20/15 | Tue 12/1/15 |
| 6 | | | Analysis+Prototyping | 40 days | Mon 11/2/15 | Fri 12/25/15 |
| 7 | | | PDR | 2 days | Thu 11/12/15 | Fri 11/13/15 |
| 8 | | | Prototype 1 | 25 days | Fri 1/1/16 | Thu 2/4/16 |
| 9 | | | Finish CAD | 7 days | Mon 1/4/16 | Tue 1/12/16 |
| 10 | | | CAM/Machine Part | 14 days | Mon 1/11/16 | Thu 1/28/16 |
| 11 | | | Assembly | 5 days | Sat 1/30/16 | Thu 2/4/16 |
| 12 | | | CDR Prep | 7 days | Wed 1/27/16 | Thu 2/4/16 |
| 13 | | | CDR | 10 days | Thu 2/4/16 | Wed 2/17/16 |
| 14 | | | Testing | 23 days | Mon 2/1/16 | Tue 3/1/16 |
| 15 | | | Vibrations Test | 12 days | Mon 2/15/16 | Mon 2/29/16 |
| 16 | | | 4th Axis | 41 days | Tue 3/1/16 | Sun 4/24/16 |
| 17 | | | Parts Ordering | 9 days | Tue 3/1/16 | Thu 3/10/16 |
| 18 | | | CAM 2 | 6 days | Tue 3/1/16 | Mon 3/7/16 |
| 19 | | | Manufacturing | 21 days | Tue 3/1/16 | Mon 3/28/16 |
| 20 | | | Shaft | 2 days | Fri 3/4/16 | Sat 3/5/16 |
| 21 | | | Platter | 3 days | Mon 3/7/16 | Wed 3/9/16 |
| 22 | | | Preload | 4 days | Wed 3/9/16 | Sat 3/12/16 |
| 23 | | | Housing | 6 days | Mon 3/14/16 | Sun 3/20/16 |




Project: Project1
Date: Thu 6/2/16

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|---|---|---|
| Task | Inactive Summary | External Tasks |
| Split | Manual Task | External Milestone |
| Milestone | Duration-only | Deadline |
| Summary | Manual Summary Rollup | Progress |
| Project Summary | Manual Summary | Manual Progress |
| Inactive Task | Start-only | Slippage |
| Inactive Milestone | Finish-only | |

| ID | | Task Mode | Task Name | Duration | Start | Finish |
|---|---|---|---|---|---|---|
| 24 | ✓ | | Cap | 5 days | Sun 3/20/16 | Thu 3/24/16 |
| 25 | | | Assembly 2 | 26 days | Fri 3/18/16 | Fri 4/22/16 |
| 26 | | | Testing 2 | 22 days | Mon 4/4/16 | Sat 4/30/16 |
| 27 | ✓ | | Vibrations Test | 5 days | Tue 4/5/16 | Mon 4/11/16 |
| 28 | | | Motor & Accuracy Test | 6 days | Mon 4/25/16 | Sat 4/30/16 |
| 29 | | | Does it work??? | 6 days | Mon 4/25/16 | Sat 4/30/16 |
| 30 | | | Software Integration | 32 days | Mon 3/14/16 | Mon 4/25/16 |
| 31 | ✓ | | 5th Axis | 20 days | Sun 4/24/16 | Wed 5/18/16 |
| 32 | ✓ | | CAD 3 | 3 days | Sun 4/24/16 | Tue 4/26/16 |
| 33 | ✓ | | CAM 3 | 5 days | Tue 4/26/16 | Sat 4/30/16 |
| 34 | ✓ | | Parts Ordering | 2 days | Mon 4/25/16 | Tue 4/26/16 |
| 35 | ✓ | | Manufacturing 3 | 16 days | Mon 4/25/16 | Fri 5/13/16 |
| 36 | ✓ | | Shaft | 2 days | Mon 5/2/16 | Tue 5/3/16 |
| 37 | ✓ | | Adapter Plates | 8 days | Wed 5/4/16 | Fri 5/13/16 |
| 38 | ✓ | | Preload | 2 days | Sun 5/8/16 | Mon 5/9/16 |
| 39 | ✓ | | Cap | 6 days | Mon 5/2/16 | Sat 5/7/16 |
| 40 | ✓ | | Housing | 16 days | Mon 4/25/16 | Fri 5/13/16 |
| 41 | | | Assembly 3 | 4 days | Sat 5/14/16 | Wed 5/18/16 |
| 42 | ✓ | | Testing 3 | 6 days | Thu 5/19/16 | Thu 5/26/16 |
| 43 | ✓ | | Motor & Accuracy Test | 4 days | Thu 5/19/16 | Tue 5/24/16 |
| 44 | ✓ | | Does it work??? | 6 days | Thu 5/19/16 | Thu 5/26/16 |
| 45 | ✓ | | Hardware/Software | 9 days | Mon 5/16/16 | Thu 5/26/16 |
| 46 | ✓ | | Poster | 5 days | Mon 5/23/16 | Fri 5/27/16 |

/13 9/27 10/11 10/25 11/8 11/22 12/6

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Project: Project1
Date: Thu 6/2/16

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| Task | Inactive Summary | External Tasks |
| Split | Manual Task | External Milestone |
| Milestone | Duration-only | Deadline |
| Summary | Manual Summary Rollup | Progress |
| Project Summary | Manual Summary | Manual Progress |
| Inactive Task | Start-only | Slippage |
| Inactive Milestone | Finish-only | |

| ID | | Task Mode | Task Name | Duration | Start | Finish |
|---|---|---|---|---|---|---|
| 47 | | | SENIOR EXPO | 1 day | Thu 5/26/16 | Thu 5/26/16 |
| 48 | | | Final Report | 5 days | Sat 5/28/16 | Thu 6/2/16 |

/13 9/27 10/11 10/25 11/8 11/22 12/6
T M S S F T W T M S S F T W T

Project: Project1
Date: Thu 6/2/16

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| Task | Inactive Summary | External Tasks |
| Split | Manual Task | External Milestone |
| Milestone | Duration-only | Deadline |
| Summary | Manual Summary Rollup | Progress |
| Project Summary | Manual Summary | Manual Progress |
| Inactive Task | Start-only | Slippage |
| Inactive Milestone | Finish-only | |


Project: Project1
Date: Thu 6/2/16

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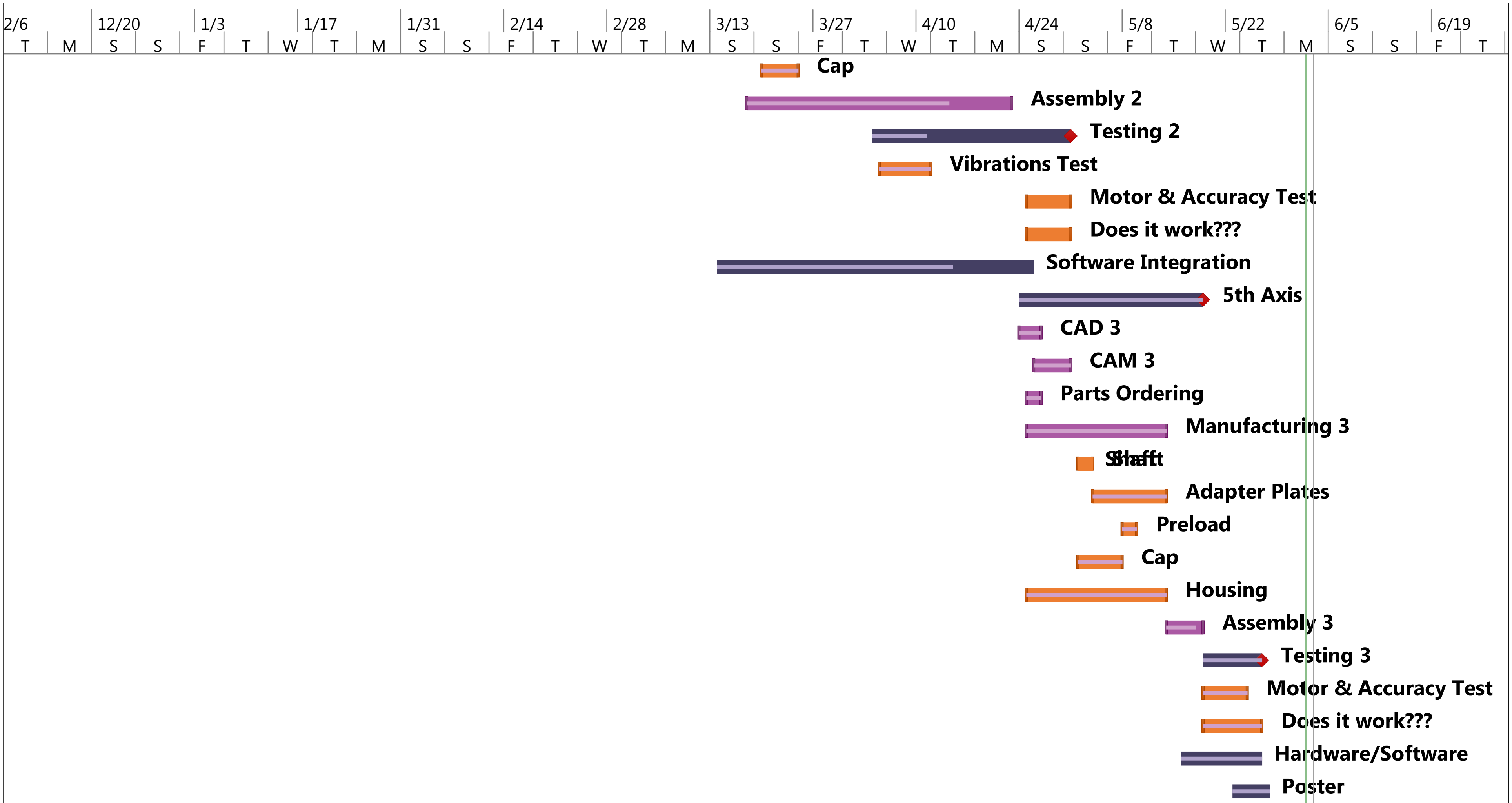
|---|---|---|---|---|---|---|---|---|
| Task | | Inactive Summary | | External Tasks | |
| Split | | Manual Task | | External Milestone | |
| Milestone | | Duration-only | | Deadline | |
| Summary | | Manual Summary Rollup | | Progress | |
| Project Summary | | Manual Summary | | Manual Progress | |
| Inactive Task | | Start-only | | Slippage | |
| Inactive Milestone | | Finish-only | | | |




Project: Project1
Date: Thu 6/2/16

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|---|---|---|
| Task | Inactive Summary | External Tasks |
| Split | Manual Task | External Milestone |
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| Summary | Manual Summary Rollup | Progress |
| Project Summary | Manual Summary | Manual Progress |
| Inactive Task | Start-only | Slippage |
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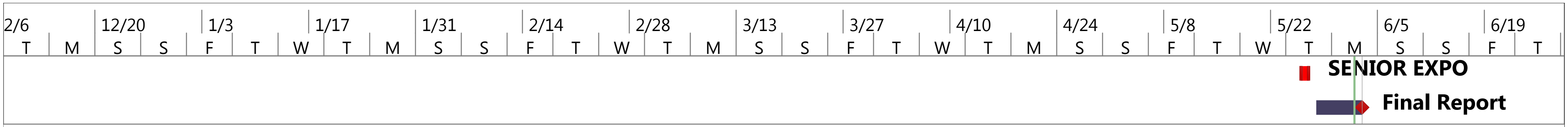


Project: Project1
Date: Thu 6/2/16

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|---|---|---|
| Task | Inactive Summary | External Tasks |
| Split | Manual Task | External Milestone |
| Milestone | Duration-only | Deadline |
| Summary | Manual Summary Rollup | Progress |
| Project Summary | Manual Summary | Manual Progress |
| Inactive Task | Start-only | Slippage |
| Inactive Milestone | Finish-only | |

# Appendix H

Operator’s Manual

**Operator’s Manual**

**To Install the Rotary Table**

1. Supporting the B-axis with one hand and holding the A-axis with the other, lift the rotary table onto the mill table. **NOTE:** Be careful when placing the rotary on the mill table to avoid pinching.
2. Use a 1-2-3 Block to support the bottom of the B-axis and to prevent the rotary from tipping.
3. Align the rotary table with the T-slots on the mill table.
4. Using toe clamps secure the rotary table to the mill table.

**Connecting the Rotary Table to the Controller**

1. Select two connectors with the same color
2. Push in the connectors being sure to align the keyway, and twist on the cover.

**Connecting the Controller to the Machine**

1. Connect power cord of controller to a 120V outlet.
2. Turn on the controller to ensure that the fan works.
3. Plug in the serial cable into the serial port on the CNC machine.